Comparative thermoeconomic analyses and multi-objective particle swarm optimization of geothermal combined cooling and power systems

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Abstract

Comparative parametric and multi-objective optimization analyses of three novel geothermal systems are performed for combined cooling and power generation. The first (Configuration (a)) consists of an absorption power cycle and an ejector refrigeration cycle, the second (Configuration (b)) of a modified Kalina cycle and an absorption refrigeration cycle, and the third (Configuration (c)) of a double-flash power cycle and an ejector refrigeration cycle, in all cases for power generation and cooling, respectively. Both thermodynamic (energy, exergy) and economic criteria are compared to gain an understanding of the characteristics and performance of these systems, and to ascertain the most appropriate system for different scenarios. Results from the parametric study show that Configuration (a) has the highest power output and exergy efficiency, but lowest cooling capacity and overall (power plus cooling) thermal efficiency, while Configuration (b) has the highest cooling capacity and thermal efficiency, but lowest power output and exergy efficiency. From an exergoeconomic perspective, Configuration (a) has the lowest and Configuration (b) the highest total specific cost. Configuration (c) maintains, generally, a thermoeconomic performance in-between those of the other two systems. The optimization results indicate that if the thermal efficiency and total specific cost are considered competing objectives over a range of well conditions, the optimal solutions obtained by the LINMAP method for Configurations (a) to (c) have thermal efficiencies of 19.1%, 43.0%, 42.4%, exergy efficiencies of 57.6%, 23.6%, 33.1%, total cost rates of 436 \$/h, 558 \$/h, 596 \$/h, and total specific costs of 29.7 \$/GJ, 66.9 \$/GJ, 43.5 \$/GJ. If the exergy efficiency and total cost rate are considered competing objectives, the corresponding values are 13.0%/29.1%/10.5%, 67.3%/30.5%/37.3%, 362/353/384 \$/h, and 24.9/67.5/42.7\$/GJ, respectively.

Keywords: absorption power, combined cooling and power, ejector refrigeration, geothermal, multi-objective optimization, thermoeconomic analysis

Nomenclature			
A	Area, m ²	CCP	Combined cooling and power
c	Specific exergy cost, \$/J	CHP	Combined heat and power
$\dot{C}$	Cost rate, \$/h	CI	Capital investment
d	Borehole depth of geothermal well, m	com	Compressor
D	Diameter, m	con	Condenser
f	Exergoeconomic factor	CRF	Capital recovery factor
$\dot{E}$	Exergy rate, W	crit	Critical
ED	Euclidean distance	CS	Cooling set
h	Specific enthalpy, J/kg	D	Destruction
i	Interest rate	diff	Diffuser
$\dot{m}$	Mass flow rate, kg/s	eje	Ejector
n	Number of operating years	EV	Expansion valve
P	Pressure, Pa	eva	Evaporator
PR	Pressure ratio	F	Fuel
$\dot{Q}$	Heat transfer rate, W	geo	Geothermal
R_C	Relative closeness	GWP	Global warming potential
s	Specific entropy, J/kg K	HE	Heat exchanger
T	Temperature, °C or K	in	Inlet
U	Heat transfer coefficient, W/m ² K	ip	Inlet pipe
V	Volume, m ³	is	Isentropic
w	Weighting coefficient	k	k^{th} component
$\dot{W}$	Power, W	L	Loss
x	Quality	mix	Mixer
X	Mass fraction, %	MF	Mixed fluid
Z	Investment cost, \$	MOPSO	Multi-objective particle swarm optimization
$\dot{Z}$	Cost rate of components, \$/h	N	Normalized
Z_{drill}	Drilling cost, \$	noz	Nozzle
Z_{SI}	Surface installation cost, \$	ODP	Ozone depletion potential
$\dot{Z}^{\text{CI}}$	Capital investment cost rate of components, \$/h	OM	Operating and maintenance

$\dot{Z}^{OM}$	Operating and maintenance cost rate of components, \$/h	out	Outlet
<i>Greek letters</i>		pu	Pump
ε	Heat exchanger efficiency	P	Product
ζ	Geothermal gradient, °C/m	PP	Pinch point
η_{ex}	Exergy efficiency, %	PF	Primary fluid
η_{th}	Thermal efficiency, %	Q	Heat
η	Efficiency, %	ref	Reference
θ	Correction coefficient	SHE	Solution heat exchanger
ρ	Density, kg/m ³	SI	Surface installation
τ	Annual plant operation hours, h	SF	Secondary fluid
φ	Maintenance factor	SP	Separator
<i>Subscripts, superscripts and abbreviations</i>		tu	Turbine
0	Dead state	tot	Total
abs	Absorber	va	Valve
b	Booster	VG	Vapour generator
CEPCI	Chemical Engineering Plant Cost Index	W	Work

1 Introduction

Geothermal energy has some crucial advantages relative to other renewable energy sources, including primarily its capacity to drive continuous system operation with high capacity factors, reduced operational variability and need for storage. Geothermal cogeneration systems have appeared as particularly promising alternatives to fossil-fuel power plants in this context, owing to their unique potential to deliver multiple useful energy vectors at higher overall efficiencies than separate, standalone systems, and been studied extensively. Specific interest, especially in global regions with warmer climates, exists in the cogeneration of combined cooling and power (CCP).

Many researchers have investigated conventional geothermal plants and proposed novel geothermal systems based on various cycles and configurations thereof [1]. The selection of a suitable working fluid is of primary importance in organic Rankine cycle (ORC) systems, which is a particularly popular cycle in these applications, leading Zare [2] to consider different fluids for such systems when utilizing medium-grade geothermal brine. Isobutane was identified as a particularly suitable working fluid in this study when considering both thermodynamic and economic performance indicators. In another study, Altun and Kilic [3] evaluated the thermodynamic performance of a geothermal ORC system and reported that the exergy efficiency of the system can be as high as 39% if this is designed appropriately.

Other cycles and configurations have also been proposed for use in geothermal applications, *e.g.*, single- and double-flash binary cycles. Yari [4] evaluated different geothermal power plants considering various configurations based on energy and exergy analyses. The author concluded that the double-flash binary system can in some cases show improved performance relative to conventional ORC systems, in terms of thermal and exergy efficiencies. In addition to ORC and flash cycles, Kalina systems have also shown comparable performance in applications involving lower-grade geothermal sources. In this regard, Fiaschi *et al.* [5] assessed the performance of a Kalina-based geothermal system and compared the performance of such a system to equivalent ORC systems. They reported that Kalina systems can produce 22-42% more power compared to ORC systems, although this conclusion does not include considerations of system cost or financial performance. Furthermore, Shokati *et al.* [6] showed that LiBr-water absorption power cycles can outperform conventional Rankine cycles in terms of efficiency and power output, *i.e.*, again in the absence of economic considerations. In any case, from studies such as these, there

is evidence that LiBr-water absorption power cycles can be employed as a promising alternative to conventional Rankine and Kalina systems when geothermal sources are used.

Of particular interest to the present research are multigeneration geothermal systems. Such systems have important advantages when used in residential and industrial applications, due to the higher overall energy generation and their ability to cater to the holistic energy demands of end-users, which invariably involve multiple energy vectors. Van Erdeweghe [7] compared the performance of a range of different low-grade geothermal combined heat and power (CHP) systems with ORC technology for heat-to-power conversion. The results from this study indicated that the second-law efficiency of such CHP systems can be up to 23% higher than that of conventional (power-generating only) geothermal plants. Similarly, Kalina cycle systems have shown great functionality after being integrated with geothermal systems for multigeneration purposes. Zare [8] compared the performance of trigeneration systems featuring Kalina cycles when utilizing low-grade geothermal sources from a thermodynamic standpoint. Results revealed that for a low-temperature geothermal brine temperature of 120 °C, the Kalina cycle-based system might produce 12% higher electricity compared to conventional ORC systems.

Among multigeneration systems, geothermal CCP systems are prominent in industrial and residential applications. Motivated by the knowledge that Kalina cycles have shown promising performance when utilizing low-grade heat sources, Cao *et al.* [9] investigated a Kalina system internally integrated with an absorption refrigeration system driven by low-grade geothermal heat source for cooling and power. This work showed that the second-law efficiency and power generation capability of this system can exceed that of standalone systems. Zhao *et al.* [10] investigated another type of geothermal CCP system, this time incorporating an ORC system and an ejector refrigeration cycle through internal integration and indicated that ejectors are suitable for flexible integration with ORC systems. The aforementioned studies focused on the thermodynamic performance of geothermal CCP systems, without economic considerations, but other studies have investigated the systems from a thermoeconomic standpoint, including a Kalina [11] and a geothermal cascade system [12]. Fiaschi *et al.* [5], for example, investigated the economic performance of Kalina- and ORC-based systems driven by geothermal heat sources. These authors reported that the specific cost of electricity associated with ORC systems is a little lower (by 3%), which aligns with knowledge in other applications of these two technologies. In all cases, the heat-source temperature is identified as a crucial parameter that affects the thermodynamic and economic

performance of such systems. Accordingly, the impact of the geothermal heat source on the thermoeconomic performance of binary-cycle geothermal plants was assessed by Liu *et al.* [13], who concluded that an elevated geothermal brine temperature, leads to an increase in all objective functions, *i.e.*, capital cost, power generation, and second law efficiency.

Even though numerous studies have been performed on proposing and investigating multigeneration geothermal systems from exergetic [14], economic [15] and thermal/thermoeconomic [16] standpoints, less research has been dedicated to optimizing the proposed multigeneration and/or conventional geothermal systems. Nevertheless, some excellent studies have been published in this space, especially more recently. Van Erdeweghe *et al.* [17] performed design optimization of different system configurations in off-design operation and reported improved economics when using multiple configurations as well as improved geothermal energy-source utilization relative to electrical power plants. Based on their results, they described an optimal binary geothermal CHP system configuration. In another similar study [18], the same authors optimized geothermal-based ORC systems while considering off-design operation and performance. They found that a crucial parameter that significantly affects the performance of such systems is the turbine inlet pressure; crucial parameters such as this should be considered when such systems are optimized. Furthermore, optimization which takes into account economic criteria is increasingly necessary as economic issues are important to many stakeholders. For this reason, thermoeconomic optimization with interesting findings was considered in Ref. [19] for a flash binary geothermal system and further in Ref. [20] for various flash binary enhanced geothermal system configurations.

Nevertheless, few studies to date have focused on the optimization of geothermal systems using a multi-objective optimization [21]. Of particular relevance here is the work of Clarke and McLeskey [22], who conducted multi-objective optimization of binary geothermal power-plant systems and concluded that multi-objective swarm optimization is a preferred approach to other algorithms for the optimization of these systems. Even less attention has been paid to the multi-objective optimization of geothermal multigeneration systems [23]. On the other hand, although multi-objective optimization of such systems considering various objective functions, *e.g.*, efficiency and cost, can help identify suitable system designs for different decision-makers, only a handful of studies have undertaken such multi-objective optimization of geothermal systems

considering both thermodynamic and thermoeconomic viewpoints focusing on the selection of working fluids [24] and conventional geothermal systems [25].

Despite the wide interest and significant work overall into geothermal energy systems, multi-objective optimization studies of geothermal multigeneration systems are scarce in the literature. To the best of the authors' knowledge, most of the research reported in the literature relates to energy and exergy analyses of multigeneration geothermal systems, with few studies focusing on economic criteria and overall system optimization with respect to a variety of technical or economic performance indicators or metrics. It is noteworthy, in particular, that almost no attention has been paid to the multi-objective optimization of multigeneration geothermal systems considering thermodynamic and thermoeconomic aspects.

In this study, three novel cogeneration geothermal systems are proposed and then analysed comprehensively from both thermodynamic and economic perspectives, extending to optimizing the proposed systems by implementing the multi-objective particle swarm optimization in order to maximize a selection of objective functions while minimizing cost.

In summary, the primary objectives of the present study are:

- to propose and investigate comprehensively three novel geothermal CCP systems, each based on different technologies and configuration;
- to apply energy, exergy and thermoeconomic analyses to the proposed configurations;
- to provide side-by-side comparisons of the performance of the systems, as well as comparisons to competing conventional systems, where possible; and
- to optimize the proposed systems using multi-objective particle swarm optimization methods with respect to competing objective functions;

2 Selection and description of proposed system configurations

Absorption power cycles are power generation systems in which a vapour generator, heat exchanger, expansion valve, absorber, pump and turbine are employed (see Fig. 1(a), and Fig. 2(a) excluding the components covered by dashed lines) [26], whereas Kalina cycle systems (see Fig. 1(b), and Fig. 2(b) excluding the components covered by dashed lines) [27] and double-flash binary cycle systems are more conventional power generation systems (see Fig. 1(c), and Fig 2(c)

excluding the components covered by dash lines) [4]. According to the literature, absorption power [6], Kalina [8] and double-flash binary [4] systems are the preferred and most widely considered technologies employed when low- to medium-grade geothermal heat sources are to be utilized (via conversion) for power generation, along with ORC systems.

Furthermore, absorption chiller [12] and ejector refrigeration cycle [10] systems are conventional cooling systems that can be either internally or externally combined with various power generation systems (see the cycles covered by dashed lines in Fig. 1). Ejector refrigeration cycles have shown good performance when low temperature waste heat is available [28], and are an excellent choice for internal integration with absorption power or double-flash binary systems wherein an (internal) low-temperature heat source can be provided for the ejector cooling system. Similarly, it has been reported that absorption cooling systems can be combined with other power generation cycles when sufficient waste heat is available [9], so a portion of the surplus heat from the vapour generator in a Kalina system can be used to drive an absorption chiller system, thus simultaneously generating power and supplying cooling. Ejector refrigeration cycles are a less suitable choice for integration with Kalina systems as sufficient waste heat is available in this case, for which absorption chiller systems have been reported to have better suitability [29].

Schematics of the three proposed geothermal CCP systems are shown in Fig. 1, and corresponding thermodynamic cycles associated with the proposed configurations are presented in Fig. 2. The first system configuration (Configuration (a)) consists of an absorption power system coupled to an ejector refrigeration cycle. Geothermal brine is fed to Vapour Generator 1 (Stream 1), where it heats a LiBr-water mixture, generating steam that passes to the turbine for power generation (Stream 10) and a hot (weak) LiBr-water mixture solution (Stream 6) that is sent to Vapour Generator 2, which is used in the ejector refrigeration system. Downstream of this, the weak solution flows through a solution heat exchanger (Stream 7), Expansion Valve 1 (Stream 8), and finally enters the absorber (Stream 9) along with steam from the outlet of the turbine (Stream 11). Strong LiBr-water solution after the absorber is sent to the pump (Stream 3), the solution heat exchanger (Stream 4), and back to Vapour Generator 1 to be boiled by the geothermal brine. A refrigerant in the ejector system is heated by the hot strong LiBr-water solution at the Vapour Generator 2 (Stream 15), before entering the ejector as the primary fluid (Stream 16). The refrigerant leaves the evaporator toward the booster, which is employed to increase pressure and enhance cooling performance (Stream 23).

The compressed fluid is drawn in by the ejector where it mixes with the primary fluid (Stream 24). The mixed fluid is then sent to the condenser (Stream 17), and a portion of it, after passing through Expansion Valve 2 (Stream 19), is sent to the evaporator for cooling (Stream 20). The remaining portion enters the pump for circulation back to the Vapour Generator 2 (Stream 14).

Configuration (b) comprises modified Kalina and absorption refrigeration cycles for power generation and cooling. Saturated NH_3 -water (Streams 4 and 5) is pumped to the solution heat exchanger and then the vapour generator (Stream 6). From there, ammonia vapour enters a condenser (Stream 17), passes via an expansion valve (Stream 18) to the evaporator where it absorbs heat and delivers cooling (Stream 19). The ammonia then mixes with weak NH_3 -water solution in the absorber (Stream 20). The fluid leaving the vapour generator (Stream 7) enters the heat exchanger where heat from the geothermal brine is utilized (Streams 1 and 2). The two-phase flow (Stream 8) is then separated; the vapour phase (strong ammonia mixture) is sent to the turbine (Stream 9), and the liquid phase (weak mixture) enters the solution heat exchanger (Stream 11). The mixed fluid phase leaving the turbine and Expansion Valve 1 (Streams 10, 13 and 14) passes through the condenser and is sent back to the absorber (Streams 14, 15 and 16).

Configuration (c) employs an ejector refrigeration cycle. Geothermal steam passes from an expansion valve to Separator 1 (Streams 1 and 2), after which the liquid phase flows through a second valve to Separator 2 (Streams 4 and 5) while the steam is sent via the turbine (Streams 3 and 6) to a condenser (Stream 7). The liquid from Separator 2 enters the ejector as the primary flow (Stream 9), entraining the secondary flow (Stream 15). The two-phase flow exiting the ejector enters a second condenser where it rejects heat, and is split (Stream 10); a portion is sent to the reinjection well via a pump (Stream 16), and the rest through Expansion Valve 3 (Stream 12) to the evaporator, where it boils to provide cooling (Stream 13), the compressor, and finally the ejector (Streams 14 and 15).

Figure 1: Proposed combined cooling and power (CCP) geothermal systems

Figure 2: Thermodynamic cycles of the proposed system configurations on T - s diagrams

3 Methods

The results from the assessments of the proposed CCP systems that are presented in this paper are based on thermodynamic (energy and exergy) and economic analyses. The EES software environment was used to solve thermodynamic relations, along with equations describing mass, energy, exergy and cost conservation. The proposed systems were then optimized by linking EES to MATLAB codes, and using a particle swarm optimization method. The detailed procedure and corresponding mathematical modelling are described in this section.

The following assumptions have been made in this work:

- steady-state conditions are assumed throughout;
- pressure drops within pipework are neglected;
- heat losses from all components to the ambient are neglected;
- kinetic and potential energy changes are neglected;
- outlets of all condensers, separators and evaporators are at saturated states;

and the models (including the assumptions) are validated against data in the literature in Section 4.1.

3.1 Thermodynamic analysis

First- (energy) and second-law (exergy) analyses were applied to each of the proposed system configurations using steady-state energy and exergy balance equations [30]:

$$\sum(\dot{m}h)_{\text{in}} - \sum(\dot{m}h)_{\text{out}} + \sum \dot{Q}_{\text{in}} - \sum \dot{Q}_{\text{out}} - \dot{W} = 0 \quad (1)$$

$$\sum \dot{E}_{\text{in}} = \sum \dot{E}_{\text{out}} + \dot{E}_{\text{D}} \quad (2)$$

in which $\dot{m}$, h , $\dot{Q}$, $\dot{W}$, and $\dot{E}$ refer to mass flow rates, enthalpies, heat flow rates (or, heating loads), power (done by the system), and exergy rates. Subscripts ‘in’, ‘out’ and ‘D’ denote inlets, outlets, and destruction in the case of exergy rate. These two equations are applied to all components of the systems investigated in this work. The full list of component parameter values is given in Table 1.

The ejector is relatively more complicated than the other components in the systems under consideration in this work, so a brief description of the modelling of this component is provided here. The modelling of the ejector was conducted by following the methodology employed by Dai *et al.* [31]. This method is based on a one-dimensional (1-D) constant pressure approach that is widely used for ejector modelling, the fundamentals of which were developed by Huang *et al.* [32]. In addition, frictional losses were taken into account as neglecting such losses has been shown to reduce the reliability of the results [33]. The following assumptions, listed below, have been made specifically with regard to the modelling of this component [31]:

- 1-D and steady-state conditions are assumed for the flow throughout the ejector;
- fluid velocities at the inlet and outlet of the ejector are ignored;
- frictional losses are accounted for via appropriate nozzle, diffuser and mixing section efficiencies;
- constant pressure is considered in the mixing section;

- the ejector is insulated, so heat losses from the outer walls to the ambient are negligible.

After solving the set of energy balance equations throughout each CCP system configuration, including simulating the flow and thermodynamic processes through the ejector where relevant, the net work (in fact, power) output and cooling capacities/loads can be calculated for each case. An overall (cooling plus power) thermal efficiency of each of the proposed systems can then be calculated relative to the thermal energy taken from the geothermal heat source [4]:

$$\eta_{th} = \frac{\dot{W}_{net} + \dot{Q}_{eva}}{\dot{m}_1(h_{geo,in} - h_{geo,out})} \quad (3)$$

in which $\dot{Q}_{eva} = \dot{m}_{21}(h_{22} - h_{21})$ [34], and the subscripts ‘eva’, ‘geo,in’ and ‘geo,out’ denote the evaporator, the geofluid inlet and the geofluid outlet, respectively.

Furthermore, an alternatively overall thermal efficiency definition is considered relative to the total thermal energy available in the geothermal heat source [4]:

$$\eta'_{th} = \frac{\dot{W}_{net} + \dot{Q}_{eva}}{\dot{m}_1(h_{geo,in} - h_0)} \quad (4)$$

where the subscript ‘0’ denotes the dead state. Similarly, an exergy efficiency can be evaluated considering the exergy taken from the geothermal heat source [4]:

$$\eta_{ex} = \frac{\dot{W}_{net} + \dot{E}_{eva}}{\dot{E}_{geo,in} - \dot{E}_{geo,out}} \quad (5)$$

where $\dot{E}_{eva} = \dot{E}_{22} - \dot{E}_{21}$ [34], and an alternative exergy efficiency is also defined as [4]:

$$\eta'_{ex} = \frac{\dot{W}_{net} + \dot{E}_{eva}}{\dot{E}_{geo,in}} \quad (6)$$

The first set of definitions is concerned with how effectively the systems can operate relative to the fractional energy/exergy taken from the heat source, whereas the second set of definitions is concerned with the effectiveness of the systems relative to the total (maximum) energy/exergy available in the heat source, and result in lower efficiencies compared to the former. In most results presented in this paper, specifically all results in Sections 4.3 to 4.5, Eqs. (3) and (5) are used as the primary efficiency definitions of interest, however, results from those definitions to results obtained with alternative efficiency definitions (*i.e.*, Eqs. (4) and (6)) in Section 4.6 are compared as well. The input conditions and all parameter values used for the simulation of all three CCP systems investigated in this work are summarized in Table 1.

Table 1: Input parameters for the simulation of the proposed systems

Parameter	Value	Parameter	Value
All configurations			
Ambient temperature, T_0	20 °C	Ambient pressure, P_0	101.3 kPa
Borehole geothermal depth, d	3800 m [35]	Geothermal fluid temperature, T_{geo}	See Appendix A
Geothermal brine flow rate, $\dot{m}_1$	100 kg/s [36]	Geothermal fluid pressure, P_1	750 kPa
Nozzle and diffuser efficiencies of the ejectors, $\eta_{\text{noz}} = \eta_{\text{diff}}$	0.85 [37]	Temperature of the condensers and absorbers, $T_{\text{con}}, T_{\text{abs}}$	35 °C [36]
Turbines efficiency, η_{tu}	0.85 [36]	Mixing section efficiency of the ejectors, η_{mix}	0.9 [37]
Pumps efficiency, η_{pu}	0.85 [38]	Evaporators temperature, T_{eva}	7 °C
Configuration (a)			
Temperature difference between Streams 1 and 10, ΔT_{VG1}	10 °C [39]	Turbine inlet pressure, $P_{\text{in,tu}}$	100 kPa
Vapour Generator 2 pressure, P_{VG2}	1300 kPa	LiBr mass fraction of Stream 3, X_{LiBr}	36%
Temperature difference between Streams 2 and 5, $\Delta T_{\text{VG1,hot}}$	3 °C	Pressure ratio of the booster, PR_{b}	1.6 [39]
Efficiency of the solution heat exchanger, ε_{SHE}	0.83 [40]	Pinch point temperature difference in the evaporator and Vapour Generator 2, $\Delta T_{\text{PP-VG2}}, \Delta T_{\text{PP-eva}}$	3 °C
Configuration (b)			
Temperature difference between Streams 2 and 17, ΔT_{VG}	10 °C [39]	Temperature difference between Streams 2 and 7, $\Delta T_{\text{hot,HE}}$	10 °C [39]
Efficiency of the solution heat exchanger, ε_{SHE}	0.83 [40]	Temperature difference between Streams 1 and 2, ΔT_{HE}	15 °C
Configuration (c)			
Separator 1 pressure, P_{SP1}	150 kPa	Condenser 2 pressure, P_{con2}	5 kPa
Separator 2 pressure, P_{SP2}	50 kPa	Pressure ratio of the compressor, PR_{com}	2 [39]
Condenser 1 pressure, P_{con1}	10 kPa	Compressor efficiency, η_{com}	0.85 [39]
Exergoeconomic parameters			
Interest rate, i	0.12	Number of operating years, n	20 yr [36]
Annual operating hours, τ	330×24 h	Operating and maintenance factor, φ	1.06 [41]

3.2 Exergoeconomic analysis

Exergoeconomic approaches are useful in assessing the economics of thermodynamic systems such as those in the present study from an exergy-based (second law) perspective. In these approaches, the cost flows, total cost rates and specific costs associated with all exergy flows throughout the system are estimated, along with the cost of exergy destruction and exergoeconomic factors, which are especially important economic performance indicators.

In performing this analysis, cost balance equations were first written for each component ‘ k ’, as suggested in Ref. [42], in order to calculate the aforementioned factors:

$$\sum \dot{C}_{\text{out},k} + \dot{C}_{w,k} = \sum \dot{C}_{\text{in},k} + \dot{C}_{q,k} + \dot{Z}_k \quad (7)$$

where $\dot{C}$ denotes the cost rate, and subscripts ‘in’, ‘out’, ‘q’ and ‘w’ refer to the inlet, outlet, heat and work.

The total cost rates of all components ($\dot{Z}_k$) and flows were then obtained, respectively, from [42]:

$$\dot{Z}_k = \dot{Z}_k^{\text{CI}} + \dot{Z}_k^{\text{OM}} \quad (8)$$

$$\dot{C}_m = c_m \dot{E}_m \quad (9)$$

for each component ‘ k ’, where superscript ‘CI’ refers to capital investment, ‘OM’ to operating and maintenance expenses, and subscript ‘ m ’ can be replaced by ‘in’, ‘out’, ‘w’ or ‘q’.

Furthermore, the cost functions $\dot{Z}_k^{\text{CI}}$ and $\dot{Z}_k$ were calculated from [42]:

$$\dot{Z}_k^{\text{CI}} = \left(\frac{\text{CRF}}{\tau} \right) Z_k \quad (10)$$

$$\dot{Z}_k = \dot{Z}_k^{\text{CI}} \times \varphi \quad (11)$$

in which $\dot{Z}_k$ is the total cost rate of component ‘ k ’, including capital investment plus operating and maintenance cost rates. In addition, τ represents the annual operating hours excluding overhaul/repairing periods, φ is the maintenance factor, which was taken as 1.06 in this study, and the capital recovery factor (CRF) was found by considering inflation [42]:

$$\text{CRF} = \frac{i(1+i)^n}{(1+i)^n - 1} \quad (12)$$

where i is the interest rate, and n is the number of working years (20 years for most such plants) [43].

The current costs of the components were then evaluated from [30]:

$$\text{Cost of component in present year} = \frac{\text{CEPCI of present year}}{\text{CEPCI of original year}} \quad (13)$$

where the Chemical Engineering Plant Cost Index (CEPCI) adjusts values year to year.

Exergoeconomic parameters are calculated after solving the component cost balances [42,44]:

$$c_{P,k} = \frac{\dot{C}_{P,k}}{\dot{E}_{P,k}} \quad (14)$$

$$c_{F,k} = \frac{\dot{C}_{F,k}}{\dot{E}_{F,k}} \quad (15)$$

$$\dot{C}_{D,k} = c_{F,k} \times \dot{E}_{D,k} \quad (16)$$

$$\dot{C}_{L,k} = c_{F,k} \times \dot{E}_{L,k} \quad (17)$$

$$f_k = \frac{\dot{Z}_k}{\dot{Z}_k + \dot{C}_{D,k} + \dot{C}_{L,k}} \quad (18)$$

where $c_{P,k}$, $c_{F,k}$, $\dot{C}_{P,k}$, $\dot{C}_{F,k}$, $\dot{C}_{D,k}$ and $\dot{C}_{L,k}$ are the unit product and fuel costs, cost rate of product and fuel, exergy destruction and exergy loss cost rates of each component, and f_k is the exergoeconomic factor of component 'k', which provides a relative comparison of the total cost rate and cost of exergy destruction and loss of that component. Further, $\dot{E}_{F,k}$ and $\dot{E}_{D,k}$ and $\dot{E}_{L,k}$ denote the rate of exergy of the products and fuels, the exergy destruction and the exergy loss for component k.

Finally, the systems were compared with respect to their exergoeconomic performance. The cost balance equations, cost functions, and auxiliary equations required to perform the comparison are listed in [Appendix A](#), where the total cost rate of each system was evaluated from:

$$\dot{C}_{\text{tot}} = \sum_{j=1}^{j=k} \dot{Z}_j \quad (19)$$

and the total specific cost of each system in \$/GJ was evaluated from:

$$c_{P,\text{tot}} = \frac{\dot{C}_{\text{tot}}}{(\dot{W}_{\text{net}} + \dot{E}_{\text{eva}}) \times 0.0036} \quad (20)$$

in which the factor 0.0036 in the nominator is added in order to convert \$/kWh to \$/GJ.

3.3 Multi-objective particle swarm optimization

All systems were optimized by considering multi-objective functions using MATLAB. The decision variables used for the optimization are listed in Table 2. It should be noted that some permutations are impossible, so they were neglected, and only possible and applicable permutations were considered for the optimization.

Table 2: Constraints imposed on decision variables for multi-objective optimization

Parameter	Constraint	Description
<i>All Configurations</i>		
d (m)	3800-5000	Heat source temperature limitation for the proposed systems, also drilling depth limitation, the corresponding heat source temperature is $125 < T_{\text{geo}} (\text{°C}) < 165$
<i>Configuration (a)</i>		
$\Delta T_{\text{VG1}} (\text{°C})$	8-15	Cold stream temperature cannot exceed the hot stream temperature in the vapour generator
P_{VG2} (kPa)	1200-1600	Vapour Generator 2 and ejector pressure limit
$P_{\text{in,tu}}$ (kPa)	50-110	Turbine inlet pressure limit which should be lower than the critical pressure
X_{LiBr} (%)	20-37	LiBr mass fraction limitation for cooling and power
PR_{b}	1.2-1.6	Booster pressure ratio limited by the cooling set and ejector
<i>Configuration (b)</i>		
$\Delta T_{\text{VG}} (\text{°C})$	6-15	Cold stream temperature cannot exceed hot stream temperature in the vapour generator
$\Delta T_{\text{HE}} (\text{°C})$	10-35	Cold stream temperature cannot exceed hot stream temperature in the heat exchanger
$\Delta T_{\text{hot,HE}} (\text{°C})$	7-20	Cold stream temperature cannot exceed hot stream temperature in the heat exchanger
<i>Configuration (c)</i>		
PR_{com}	1.4-2.8	Compressor pressure ratio limited by the cooling set and ejector
P_{SP1} (kPa)	80-300	Separator 1 and turbine pressure limit to prevent the formation of liquid droplets
P_{SP2} (kPa)	20-70	Separator 2 pressure limit which should be lower than Separator 1 pressure
P_{con1} (kPa)	8-15	Turbine outlet pressure limit to prevent the formation of liquid droplets

Multi-objective particle swarm optimization (MOPSO) was used to optimize the systems through a dynamic weighted aggregation approach [45]. Particle swarm optimization (PSO) was

preferred to genetic or other evolutionary-based algorithms, following other simulation-based optimization studies of geothermal energy systems [22]. A similar procedure to that of Clarke *et al.* [22] was used through the formulation of the PSO algorithm in MATLAB and by linking this to EES. Two sets of competing objective functions were considered: (1) total specific cost vs. thermal efficiency, and (2) total cost rate vs. exergy efficiency.

For each multi-objective optimization case, Pareto frontiers consisting of optimum solution points were extracted with three highlighted points on each one; Point A: denoting the maximum thermal efficiency, Point B: indicating the minimum total specific cost, and Point C: showing a balanced condition between efficiency and cost using LINMAP and TOPSIS methods.

Euclidean distances can be evaluated between the optimal solutions and ideal/non-ideal solution points from the LINMAP and TOPSIS methods. For example, when the thermal efficiency and total specific cost are considered as the objective functions, Euclidean distances can be calculated between the normalized optimal and ideal solutions as follows [46]:

$$ED_i^+ = \sqrt{w_1(\eta_{th,i}^N - \eta_{th,ideal}^N)^2 + w_2(c_{P,tot,i}^N - c_{P,tot,ideal}^N)^2} \quad (21)$$

where w_1 and w_2 are weighting coefficients and the normalized functions are determined from [46]:

$$\eta_{th,i}^N = \frac{\eta_{th,i}}{\sqrt{\sum_i (\eta_{th,i})^2}}, \quad c_{P,tot,i}^N = \frac{c_{P,tot,i}}{\sqrt{\sum_i (c_{P,tot,i})^2}} \quad (22)$$

The weighting coefficients are set to unity when the two objective functions are equally important, or different values can be selected when one of the objective functions is more important. In this study, equal weighting coefficient values (*i.e.*, $w_1 = w_2 = 1$) have been considered.

The lowest value of ED_i^+ in Eq. (21) was selected as the final optimum when using the LINMAP method. In addition, non-ideal solutions indicated as ED_i^- were also considered for the TOPSIS method, following a similar approach [46]:

$$ED_i^- = \sqrt{w_1(\eta_{th,i}^N - \eta_{th,non-ideal}^N)^2 + w_2(c_{P,tot,i}^N - c_{P,tot,non-ideal}^N)^2} \quad (23)$$

Subsequently, a relative closeness (R_C) was evaluated, defined as:

$$R_{C,i} = \frac{ED_i^-}{ED_i^- + ED_i^+} \quad (24)$$

where the maximum value of R_C was selected as the final optimum when using the TOPSIS method.

A similar approach was taken for the selection of final optimal points when exergy efficiency and total cost rate were considered as the objective functions. The input parameters used to conduct MOPSO and the procedure followed in this study are presented in Fig. 3.

Figure 3: Input parameters used and MOPSO procedure followed in this study

4 Results and discussion

The three proposed configurations are compared in terms of energy, exergy and exergoeconomic criteria to determine the strengths and weaknesses of each one. Firstly, results from a parametric study are presented to assess the impact of key decision variables on system performance. Subsequently, results relating to optimized systems through multi-objective optimization are presented. Prior to these assessments, the system performance and cost indicators were evaluated when employing different working fluids at the baseline (see Table 1). For Configuration (a), a wide range of hydrocarbons with low global warming and ozone depletion potentials (GWP and ODP) were considered, including Ethylbenzene, Isobutane, Isobutene, Isopentane, Isohexane, n-Butane, n-Pentane, n-Hexane, n-Octane, n-Undecane, o-Xylene, Toluene, and others, e.g.,

Isopropanol. Further useful information on these refrigerants can be found in Ref. [47]. It was concluded that Isobutane is a particularly favourable working fluid for this configuration, as it results in a well-balanced condition in terms of thermal and exergy efficiencies, and total specific cost, so this fluid is selected for the rest of this study (see Table 3). For Configurations (b) and (c), an ammonia-water mixture and water as working fluids are used, respectively.

Table 3: System performance and characteristics of working fluids examined for Configuration (a)

Fluid	η_{th} (%)	η_{ex} (%)	CP_{tot} (\$/GJ)	GWP (100 yr)	ODP (100 yr)	Safety group	T_{crit} (°C)	P_{crit} (MPa)
Isobutane	18.5	56.6	32.6	3	0	A3	135	3.64
Toluene	18.2	56.9	33.3	0	0	–	319	4.13
Isohexane	17.7	56.7	33.9	3-20	0	A3	225	3.04
Isobutene	18.5	56.8	33.5	3	0	A3	145	4.01
Ethylbenzene	17.7	57.1	33.1	–	0	–	344	3.62
Isopentane	17.2	57.2	36.9	11	0	A3	187	3.37
Isopropanol	17.0	57.6	33.6	3-20	0	–	235	4.76
n-Butane	18.4	56.7	35.4	3-20	0	A3	152	3.80
n-Hexane	17.9	56.7	33.6	3-20	0	A3	235	3.06
n-Octane	16.7	57.1	33.0	3-20	0	A3	296	2.50
n-Pentane	17.6	56.9	35.6	11	0	A3	196	3.36
n-Undecane	16.0	57.1	32.6	3-20	0	–	366	1.99
o-Xylene	17.6	57.1	33.0	3-20	0	–	357	3.74
R1243zf	15.0	56.7	32.7	0.82	0	A2L	106	3.61
R1234yf	13.4	56.8	33.2	4	0	A2L	94.7	3.38
R1234ze(E)	15.9	56.7	33.3	<1	0	A2L	109	36.4
Novec649	15.7	56.9	34.2	<1	0	–	169	1.88
Dimethylether	15.3	57.0	33.2	1	0	A3	127	5.34

4.1 Validation

The results provided in Table 4 are provided as a form of verification of the absorption and ejector refrigeration system models, and of the absorption power cycle model. For this, results from a number of recognized numerical studies in the literature have been employed, selected where it was deemed that sufficient information was provided for our purposes. Specifically, Garcia-Hernando *et al.* [48] modelled an absorption power cycle (others, *e.g.*, Novotny and Kolovratnik [49] also reported similar results) and Lostec *et al.* [50] examined an absorption refrigeration cycle experimentally, so our results are compared to those provided by these authors. Furthermore, an experimental study on the ejector refrigeration cycle [51] was selected here for the verification of this subsystem. The maximum absolute error resulting from the comparison of the present study results with those reported in these publications is 1.4% for the simulation of the absorption power cycle, and 5.0% and 4.5% for the absorption and ejector refrigeration systems, respectively.

Table 4: Verification of absorption power, absorption refrigeration and ejector refrigeration subsystems, and of the system geothermal section

<i>Absorption power cycle (steam generator temperature = 100 °C, absorber pressure = 7.4 kPa)</i>			
Generator pressure (kPa)	Efficiency (%) (Ref. [48])	Efficiency (%) (Present study)	Error (%)
30.0	7.2	7.3	1.4
40.0	8.5	8.6	1.2
50.0	9.3	9.2	-1.1
57.2	9.2	9.1	-1.1
<i>Absorption refrigeration cycle (working fluid: NH₃-water, refer to Ref. [50] for full details of tests)</i>			
Test number	COP (Ref. [50])	COP (Present study)	Error (%)
Test 1	0.60	0.62	+3.3
Test 8	0.20	0.19	+5.0
Test 15	0.11	0.11	0
Test 18	0.30	0.29	-3.3
<i>Ejector refrigeration cycle (condenser temperature: 303.15 K)</i>			
Generator temperature (°C)	Entrainment ratio (Ref. [51])	Entrainment ratio (Present study)	Error (%)

102	0.10	0.10	0
104	0.45	0.44	-2.2
106	0.66	0.63	-4.5
112	0.60	0.61	+1.7

Real data of drilling and installing costs of deep geothermal wells

ζ (°C/km) / real depth (km)	Real investment cost (M\$) (Ref. [35])	Estimated cost (M\$) (Present study)	Error (%)
48.3 / 3.29 ^a	20.0	20.2	+1
47.1 / 3.24 ^b	19.0	19.0	0

Real data of geothermal fluid temperatures

ζ (°C/km) / real depth (km)	Measured brine production temperature (°C) (Refs. [35,52])	Estimated brine production temperature (°C) (Present study)	Error (%)
58.0 / 4.22 ^c	215	210	-2.3
31.9 / 4.26 ^d	115	117	+1.7

^a In Landau, Germany; ^b Average data of several real geothermal projects in Upper Rhine Graben, Germany; ^c In Cooper Basin, Australia; ^d In Taufkirchen, Germany.

In order to validate the estimation of geothermal section return temperature (brine production temperature) and costs, real data reported by Weinand *et al.* [35] and Hogarth *et al.* [52] concerning established projects in different regions of Germany and Australia were considered. From the results in Table 4, it can be concluded that predictions of the return temperature of the geothermal brine as well as of the investment cost from the models in the present work are in good agreement with those reported in the literature. It should be noted that investment costs may vary significantly across projects, labour quota, regions, amongst others; however, results from our approach can be considered reasonable midway costs. The geothermal temperature gradient (ζ), geothermal brine mass flowrate and drilling depth are all considered as inputs to the models (see [Appendix A](#) for full details).

4.2 Exergy and cost flow diagrams

The three proposed geothermal system configurations are examined here from exergetic and exergoeconomic perspectives. Corresponding results are presented in the Sankey diagrams in Figs. 4 and 5, to show exergy and cost flows respectively.

According to Fig. 4, Configuration (c) is capable of extracting more exergy from the geothermal brine heat-source stream as the brine leaving the system has less exergy compared to that of the other two configurations. Nevertheless, the net power output of Configuration (a) is the highest of all three configurations. In addition, from Fig. 5, it is observed that for a constant cost rate flowing through the system from the geothermal brine stream, the cost rate associated with the turbine power output takes the highest value for Configuration (a), while the exergoeconomic factor of the turbine is highest for Configuration (b). This indicates that the share of the cost of exergy destruction is lower than the investment cost rate of the turbine for Configuration (b). All values of f , which indicates the fraction of the cost rate of exergy destruction compared to the investment cost rate of each component, are shown in Fig. 5.

Figure 4: Exergy flows (kW) within Configurations (a) to (c), from top to bottom

Figure 5: Cost flows (\$/h) within Configurations (a) to (c), from top to bottom

4.3 Parametric study

Results are presented in this section from a sensitivity analysis performed to predict the impact of important variables and parameters within reasonable variation ranges. A number of parameters are selected as effective decision variables, and a reasonable range of variation is assigned in each case to the selected parameters. Specific input decision variables of the studied configurations are considered to complement the parametric study.

4.3.1 Configuration (a) specific parameters

Some parameters can have a direct impact on the performance of Configuration (a), so their effects should be considered. The role of the turbine inlet pressure $P_{in,tu}$ is explored in Fig. 6 over the range 30 kPa to 120 kPa, which shows significantly higher cooling capacities at higher pressures due to increased working fluid flow rates through the evaporator. However, there is an optimum net power output (3.3 MW) associated with an inlet pressure of $P_{in,tu} = 70$ kPa, at which point the enthalpy drop across the turbine is maximized.

The exergy efficiency trend closely follows that of the power output, with a local maximum exergy efficiency (59%) achieved at a turbine inlet pressure of 80 kPa. On the other hand, the overall thermal efficiency increases monotonically from 10.6% to 20.8% thanks to the significant increase in the cooling capacity within the investigated range of turbine inlet pressures. In addition, at higher turbine inlet pressures the total cost rate of this system increases as the purchase cost of the turbine increases, while from a turbine inlet pressure of 100 kPa upwards, the total cost rate reduces slightly. Furthermore, as there is an optimum point in terms of power output, there is also a corresponding minimum point for the total specific cost, since the trends of the output power and total specific cost are reversed. The total cost rate does not determine the total specific cost, as the total cost changes slightly within the investigation range and the specific cost is determined by the changes in power.

The mass fraction of LiBr, X_{LiBr} , in the absorption cycle is also an important parameter that should be considered during the design of the system; the effects of this parameter are also shown in Fig. 6. A rich/strong fluid mixture, with a higher fraction of LiBr, is expected to be beneficial in terms of cooling and thermal efficiency since a higher mass fraction can increase the enthalpy difference in the evaporator and improve the cooling capacity. However, it should be noted that when increasing

the mass fraction X_{LiBr} from 0.2 to 0.4, the net power output decreases by more than 50%, from 4.0 MW to 1.8 MW, while the exergy efficiency and total cost rate follow a similar trend, *i.e.*, decrease from about 66% to 37%, and 379 \$/h to 373 \$/h, respectively. The reduced total cost rate arises from the lower cost of the turbine, which itself relates to the reduced power rating and the requirement for a smaller component. However, as the total cost rate increases and the net power output decreases, the total specific cost increases significantly, in fact, doubles from 26.3 \$/GJ to 54.1 \$/GJ, at higher mass fractions of LiBr. It is noteworthy that the rates at which the total cost rate decreases and at which the total specific cost (slope of the curves) increase sharply at the higher studied mass fractions.

Figure 6: Effect of turbine inlet and LiBr mass fraction on performance of Configuration (a)

4.3.2 Configuration (b) specific parameters

The temperature difference between Streams 1 and 2, ΔT_{HE} , in Configuration (b) is considered as the first influential factor of interest. Results in Fig. 7 show the effect of this temperature difference on key technoeconomic performance indicators, over the range $\Delta T_{HE} = 7\text{--}20\text{ }^{\circ}\text{C}$. Larger temperature differences enable a slightly improved cooling capacity (by $\sim 3\%$), while the power output reduces by about 20%, from 760 kW to 610 kW. This can be explained in terms of the lower temperatures at the turbine inlet at higher ΔT_{HE} , and the consequent reduction in the enthalpy drop across the turbine, which reduces the generated power output. Changes to the temperature difference ΔT_{HE} do not affect the exergy and thermal efficiencies; the exergy efficiency reduces slightly from 19.5% to 16.4%, while the overall thermal efficiency remains almost constant. The reduced power output and improved cooling capacity lead to a constant thermal efficiency, while a monotonically decreasing exergy efficiency can be observed at higher ΔT_{HE} . From an exergoeconomic standpoint, there is a local optimum in terms of total cost rate (362 \$/h) at the temperature difference of 13 $^{\circ}\text{C}$, which relates to the capital cost of the components. However, the total specific cost of the system consistently increases as the temperature difference is increased further, following the trend of continuous decrease in the net power output.

Conversely, as also shown in Fig. 7, the temperature difference between Streams 2 and 7, $\Delta T_{hot,HE}$, has a different impact on the performance of this system configuration. The results indicate that a larger temperature difference, within the range 10–35 $^{\circ}\text{C}$, leads to higher net power outputs from 0.5 MW to 1.3 MW, as the enthalpy of the fluid passing through the turbine increase thanks to the increased mass flow rate of Stream 9. Consequently, the exergetic efficiency of the system improves from 14.4% to 28.8%. By contrast, larger temperature differences considerably reduce the overall thermal efficiency of the system, from 49.8% to 29.2%, due to the significant reduction in cooling capacity which drops by almost 50%, from 10.7 MW to 5.4 MW.

Figure 7: Effect of temperature difference between Streams 1 and 2 (ΔT_{HE}) and temperature difference between Streams 2 and 7 ($\Delta T_{hot,HE}$) on performance of Configuration (b)

4.3.3 Configuration (c) specific parameters

The compressor pressure ratio PR_{com} can strongly affect the performance of Configuration (c). The detailed effects of this parameter are shown in Fig. 8. Employing a higher compression pressure ratio increases the cooling capacity by a factor of more than four, from 3.7 MW to 15.3 MW within the range $1.4 < PR_{com} < 2.8$ suggesting that this pressure ratio is particularly influential in terms of cooling capacity. At this same time, higher pressure ratios negatively affect the net power output, leading to significantly lower power outputs, which halve from 1.9 MW to 0.9 MW. This loss in

power arises as a consequence of the increased enthalpy of the fluid entering the ejector, which increases the cooling load of the evaporator. However, higher pressure ratios are also associated with increased electrical consumption by the compressor, which reduces the net power output.

The effects of the compression pressure ratio PR_{com} on the thermal and exergy efficiencies of Configuration (c) are shown in Fig. 8. Raising the pressure ratio has a positive influence on the overall thermal efficiency (which increases from 14.6% to 42%), but a negative influence on the exergy efficiency (which decreases 31.1% to 17.6%). These results can be understood in terms of the lower power generation and higher cooling capacity, respectively. In addition, an increase in the pressure ratio can negatively affect the exergoeconomic metrics, as the total cost rate and total specific cost both increase. Specifically, the total cost rate increases from 385 \$/h to 396 \$/h due to the cost of the compressor and other components in the cooling sub-system, and the specific cost increases from 54 \$/GJ to 97 \$/GJ, due to the increased total cost rate and reduced power output. According to Fig. 6, lower pressure ratios are preferred if exergoeconomic metrics are a priority.

One of the most important design parameters for Configuration (c) is the inlet pressure of the second separator P_{SP2} , which is set by adjusting Expansion Valve 2. As shown in Fig. 8, there is an optimum point, at $P_{SP2} = 30\text{-}35$ kPa, which maximizes performance in terms of power generation and exergy efficiency. At a pressure of 30 kPa, the enthalpy drop across the turbine reaches its highest value, leading to the maximum power output (1.8 MW). The cooling capacity on the other hand increases monotonically from 0.6 MW to 11.8 MW as the pressure increases, and as higher enthalpy is provided to the cooling sub-system. The increase in the cooling capacity improves the thermal efficiency considerably, from 6.0% to 33.9%. Furthermore, the total specific cost increases slightly (from 379 \$/h to 392 \$/h) as the pressure increases within the investigated range, due to the increased capital cost of the cooling components, while the total specific cost varies within the range 62-72 \$/GJ, exhibiting a minimum at a pressure of 35 kPa, *i.e.*, the same as that which maximizes the exergy efficiency.

Figure 8: Effect of compressor pressure ratio and Separator 2 inlet pressure on performance of Configuration (c)

4.4 Multi-objective optimization

In this section, multi-objective particle swarm optimization results are presented and discussed. Figure 9 shows trade-offs between two pairs of competing objective functions: (1) thermal efficiency vs. total specific cost, and (2) exergy efficiency vs. total cost rate. When considering thermal efficiency and total specific cost as the competing objective functions, Pareto frontiers presenting a set of optimal solution points for the proposed configurations are presented in the first (top) row, while Pareto frontiers of optimal solutions gathered when exergy efficiency and total cost rate are selected as the

objective functions are presented in the second (bottom) row of Fig. 9. Point A in all plots indicates the maximum efficiency system design, Point B the minimum total specific cost or minimum total cost rate design, and Point C the final optimum selected by either the LINMAP or TOPSIS method.

As shown in these plots, increasing any one of the efficiency measures in any one of the system configurations always leads inevitably to rising costs. A well-balanced design is expected at this point, which can be identified through such plots. The exact values of the decision variables and performance indicators are also listed in Table 5 and Table 6 for all configuration designs indicated by Points A, B, and C, and for each one of the different competing objective function cases.

Figure 9: Pareto frontiers of optimal configurations; see Table 2 for variable range constraints

Table 5: Multi-objective optimization results compared to the base case for different optimum points/configurations when thermal efficiency and total specific cost are the objective functions; see Table 2 for decision-variable ranges imposed through constraints

Config.	d (km)	ΔT_{VG1} (°C)	P_{VG2} (kPa)	$P_{in,tu}$ (kPa)	X_{LiBr} (%)	PR_b	ΔT_{VG} (°C)	ΔT_{HE} (°C)	$\Delta T_{hot,HE}$ (°C)	PR_{com}	P_{SP1} (kPa)	P_{SP2} (kPa)	P_{con1} (kPa)	η_{th} (%)	η_{ex} (%)	$\dot{C}_{tot}$ (\$/h)	$c_{P,tot}$ (\$/GJ)	$\dot{W}_{net}$ (kW)	$\dot{Q}_{eva}$ (kW)
<i>Base case</i>																			
(a)	3.8	10	1300	100	36	1.6	–	–	–	–	–	–	–	18.5	56.6	360	32.6	3040	2080
(b)	3.8	–	–	–	–	–	10	15	10	–	–	–	–	46.2	18.8	347	108	725	9680
(c)	3.8	–	–	–	–	–	–	–	–	2	150	50	10	24.1	27.7	374	58.6	1640	7650
<i>Point A (maximum thermal efficiency)</i>																			
(a)	3.8	15	1520	110	37	1.6	–	–	–	–	–	–	–	24.4	30.2	352	68.1	1365	4190
(b)	4.3	–	–	–	–	–	15	10	17	–	–	–	–	50.5	12.0	431	148	560	14430
(c)	3.8	–	–	–	–	–	–	–	–	2.8	170	70	8	57.2	11.6	387	145	360	21760
<i>Point B (minimum total specific cost)</i>																			
(a)	5.0	9	1200	110	37	1.6	–	–	–	–	–	–	–	16.6	61.3	576	23.6	6747	1555
(b)	5.0	–	–	–	–	–	15	35	7	–	–	–	–	40.1	27.0	561	58.5	2430	13500
(c)	5.0	–	–	–	–	–	–	–	–	1.4	240	68	8	19.1	43.0	582	32.8	4840	5720
<i>Point C (well-balanced condition, LINMAP)</i>																			
(a)	4.2	15	1540	110	37	1.6	–	–	–	–	–	–	–	19.1	57.6	436	29.7	4040	2330
(b)	5.0	–	–	–	–	–	15	28	7	–	–	–	–	43.0	23.6	558	66.9	2060	14910
(c)	5.0	–	–	–	–	–	–	–	–	2.8	250	70	8	42.4	33.1	596	43.5	3450	20060
<i>Point C (well-balanced condition, TOPSIS)</i>																			
(a)	4.3	9	1600	110	37	1.6	–	–	–	–	–	–	–	18.5	61.2	441	27.6	4400	2000
(b)	5.0	–	–	–	–	–	15	30	7	–	–	–	–	42.2	24.7	559	64.0	2170	14510
(c)	5.0	–	–	–	–	–	–	–	–	2.8	250	70	8	42.4	33.1	596	43.5	3450	20060

Table 6: Multi-objective optimization results compared to the base case for different optimum points/configurations when exergy efficiency and total cost rate are the objective functions; see Table 2 for decision-variable ranges imposed through constraints

Config.	d (km)	ΔT_{VG1} (°C)	P_{VG2} (kPa)	$P_{in,tu}$ (kPa)	X_{LiBr} (%)	PR_b	ΔT_{VG} (°C)	ΔT_{HE} (°C)	$\Delta T_{hot,HE}$ (°C)	PR_{com}	P_{SP1} (kPa)	P_{SP2} (kPa)	P_{con1} (kPa)	η_{th} (%)	η_{ex} (%)	$\dot{C}_{tot}$ (\$/h)	$\dot{C}_{P,tot}$ (\$/GJ)	$\dot{W}_{net}$ (kW)	$\dot{Q}_{eva}$ (kW)
Base case																			
(a)	3.8	10	1300	100	36	1.6	–	–	–	–	–	–	–	18.5	56.6	360	32.6	3040	2080
(b)	3.8	–	–	–	–	–	10	15	10	–	–	–	–	46.2	18.8	347	108	725	9680
(c)	3.8	–	–	–	–	–	–	–	–	2	150	50	10	24.1	27.7	374	58.6	1640	7650
Point A (maximum exergy efficiency)																			
(a)	3.8	10.5	1217	109	20	1.2	–	–	–	–	–	–	–	12.7	67.8	363	24.6	4090	223
(b)	3.8	–	–	–	–	–	14	35	7	–	–	–	–	29.2	30.6	353	67.5	1362	5240
(c)	5.0	–	–	–	–	–	–	–	–	1.4	203	49	8	14.8	43.4	580	32.2	4950	3220
Point B (minimum total cost rate)																			
(a)	3.8	15	1600	110	37	1.25	–	–	–	–	–	–	–	17.4	35.1	349	59.9	1580	2210
(b)	3.8	–	–	–	–	–	15	10	12	–	–	–	–	50.1	14.2	344	143	483	10800
(c)	3.8	–	–	–	–	–	–	–	–	1.45	300	30	15	3.3	6.6	353	233	405	853
Point C (well-balanced condition, LINMAP)																			
(a)	3.8	12.8	1230	110	20	1.42	–	–	–	–	–	–	–	13.0	67.3	362	24.9	4040	342
(b)	3.8	–	–	–	–	–	12.6	35	7	–	–	–	–	29.1	30.5	353	67.5	1360	5230
(c)	3.9	–	–	–	–	–	–	–	–	1.48	80	34	8	10.5	37.3	384	42.7	2470	1710
Point C (well-balanced condition, TOPSIS)																			
(a)	3.8	12.2	1210	110	20	1.36	–	–	–	–	–	–	–	12.9	67.7	362	24.8	4060	304
(b)	3.8	–	–	–	–	–	12.6	35	7	–	–	–	–	29.1	30.5	353	67.5	1360	5230
(c)	3.9	–	–	–	–	–	–	–	–	1.48	80	34	8	10.5	37.3	384	42.7	2470	1710

4.5 Comparisons and benchmarking

A comparison of the performance of the three systems is shown in Fig. 10, with respect to different objectives. All system designs correspond to the final optimal point (Point C, LINMAP), which is a balanced condition between thermal efficiency and total specific cost (first (top) row), and exergy efficiency and total cost rate (second (bottom) row). The systems are also compared to conventional ORC [2] and Kalina cycle systems [53], that are considered appropriate benchmarks. The final optimal designs that balance thermal efficiency and specific cost (top row) suggest a preference for deeper wells (higher source temperatures) than those that balance exergy efficiency and total cost (bottom row), so an ORC system with isobutane as the working fluid is considered a suitable choice for comparison with the former systems, and a Kalina cycle system with an NH_3 mass fraction of 0.9 for higher exergy efficiency a more suitable choice for comparison with the latter.

It is interesting to note, before proceeding to discuss the results in Fig. 10, that increasing the geothermal well depth from 4.2 km to 5.0 km for Configuration (a) leads to only small changes in the results (except for total cost rate), so comparisons of this system configuration with conventional ORC systems can be considered reliable even if the depth of the geothermal well is different. For a depth of $d = 5.0$ km, the total cost rate of Configuration (a) increases by 30% and reaches 570 \$/h, which is still lower than that needed for a conventional geothermal ORC system.

From Fig. 10, it can be seen that Configuration (a) has the highest power output and exergy efficiency, but the lowest cooling capacity and overall thermal efficiency of all systems; it is also the most cost efficient as it has the lowest total specific cost. On the other hand, Configurations (b) and (c) can generate the most cooling with the latter also generating reasonable power output, and at a reasonable total specific cost. Configuration (b) has the highest thermal efficiency but also total specific cost, by far, of all systems, which is an important disadvantage of this system.

It is now important to compare the proposed systems to conventional geothermal systems. From the results in the top row of Fig. 10, which are the final optimal solutions determined by the LINMAP method when thermal efficiency and total specific cost are the competing objective functions, it can be seen that Configuration (a) outperforms an optimized conventional ORC system in almost every aspect except power output and specific cost, which are similar to those of the ORC system that, however, is optimized for power generation alone without a cooling capacity. All other configurations have higher thermal efficiencies than the conventional ORC system, but also lower power outputs, exergy efficiencies and total specific costs.

As mentioned earlier, the results in the bottom row of Fig. 10, which are the final optimal solutions determined by the LINMAP method when exergy efficiency and total cost rate are the competing objective functions, are selected for the comparison with Kalina geothermal systems. The final optimal solutions suggest a preference for shallower geothermal wells in this case (unlike the top row in the same figure), so a conventional Kalina geothermal system appears as a reasonable choice for comparison purposes at $d = 3.8$ km. From these results, Configuration (a) is shown to also outperform Kalina cycle systems except for power output and specific cost, which are similar to those of the Kalina system that, however, has zero cooling capacity.

Figure 10: Comparisons of Point C (LINMAP) system design performance for Configurations (a) to (c), also including conventional ORC and Kalina-cycle systems. Top – optimal thermal efficiency and total

specific cost, with Configuration (a): $d = 4.2$ km; and Configurations (b) and (c): $d = 5.0$ km. Bottom – optimal exergy efficiency and total cost rate, with Configurations (a), (b) and (c): $d = 3.8$ - 3.9 km.

To summarize, selecting a suitable system is based on the perspective of the decision-maker(s); if maximum exergetic efficiency/power output and minimal total specific cost are the highest priority, Configuration (a) would be the preferred system. However, if the cooling capacity is a priority along with satisfactory power output, Configuration (c) is a suitable solution. On the other hand, if overall (power + cooling) thermal efficiency is the highest priority, Configuration (b) is a favourable option. The total specific cost of Configurations (a) and (c) is comparable to fossil fuel power plans, which is one of the great advantages of these configurations. The total specific cost of Configuration (b) is 35% higher than the highest specific cost of fossil fuel power plants.

In a final step, the effects of the evaporator temperature and the depth of the geothermal well on the performance of the systems are investigated, when all other decision variables are fixed to those of the final optimum point selected by the LINMAP method. Figure 11, which focuses on system designs optimized for maximum thermal efficiency and minimum total specific cost (see Table 5, Point C), shows that the exergy efficiency of Configuration (a) is higher than those of the other systems regardless of the evaporator temperature and depth of the geothermal well. However, the exergy efficiency of Configuration (c) can become higher when a deeper geothermal well is considered; however, a lower efficiency is concluded for shallower wells. Furthermore, the overall thermal efficiency of Configuration (c) varies significantly (more than the other configurations) within the explored range of parameters. The highest thermal efficiency is obtained for higher evaporator temperatures and shallower geothermal wells, while lower evaporator temperatures and/or deeper wells can reduce the thermal efficiency of this system configuration below that corresponding to Configuration (b). Evaporator temperatures above $7\text{ }^{\circ}\text{C}$ are not applicable to Configuration (c), which is a significant drawback.

The total cost rates of all configurations are comparable and vary in a similar manner with respect to the two independent parameters (evaporator temperature and well depth). Specifically, the total cost rates are generally insensitive to the evaporator temperature and depend strongly on the well depth. The total specific cost of Configuration (c) is very high for higher evaporator temperatures and shallower geothermal wells. Configuration (b), however, shows a near-constant specific cost, while the specific cost of Configuration (a) reduces with increasing the geothermal

well depth yet remains the lowest of all systems. The explanations that underpin these reported observations and trends are similar to those provided in the parametric study section.

Finally, system designs optimized for maximum exergy efficiency and minimum total specific cost are presented (Table 6, Point C). Similar observations to those already reported in the parametric analysis (not shown) apply to the optimized systems, wherein Configuration (a) has the highest exergy efficiency and total specific cost, and Configuration (b) the highest thermal efficiency. Both efficiencies, along with the specific cost, do not show strong variations within the explored parameter space. As before, the total cost rates of all configurations are comparable and exhibit the largest variation, but with almost identical trends in response to changes in the evaporator temperature and well depth. As mentioned above, these results align closely with observations discussed (and the associated explanations) in the parametric study section.

Figure 11: Effects of evaporator temperature and borehole depth of geothermal well on performance indicators of proposed systems based on final optimal designs from LINMAP (thermal efficiency and total specific cost are objective functions)

4.6 Alternative thermodynamic performance indicators

The definitions of thermal and exergy efficiencies that have been used thus far, characterize how effectively the systems perform from a thermodynamic (first- and second-law in Eqs. (3) and (5), respectively) perspective relative to the fractional energy/exergy taken from the geothermal heat source. Alternative thermodynamic performance indicators (in Eqs. (4) and (6), respectively) can be defined relative to the total (maximum) energy/exergy available in the heat source. In this section, the results obtained for these alternative efficiency definitions are compared.

A direct comparison of the efficiency measures for optimal designs of all proposed systems at Point C (LINMAP) is presented in Fig. 12. The first set of definitions is associated with higher efficiency values, as expected, and the difference between the two thermal efficiency definitions is larger than the difference between corresponding exergy efficiency definitions. Generally, the trends of all efficiency pairs are similar, however, there are a few interesting differences of note.

With respect to the thermal efficiency, firstly, the efficiency of Configuration (a) decreases monotonically with borehole depth when considering the definition that has been used so far (Eq. (3)), but shows a clear minimum at 3.9 km and increases for deeper wells when considering the alternative definition (Eq. (5)). This has important implications as it leads to different conclusions concerning the optimal depth with respect to thermal efficiency for this system configuration; shallow for Eq. (3) but deep for Eq. (5). Secondly, for very deep wells Configuration (b) can become more efficient than Configuration (c) when considering the definition in Eq. (3), but when considering the alternative definition in Eq. (5), Configuration (b) never approaches that of Configuration (c), even at the extreme depth explored in this work of 5 km.

We also observe, with respect to the exergy efficiency, that the efficiency of Configuration (a) exhibits a maximum at a well depth of 4.7 km from the perspective of the definition in Eq. (3), but increases monotonically with depth when considering the alternative definition. This, again, is important as it suggests that the optimal depth with respect to exergy efficiency for this system configuration is not at an intermediate but at a much deeper well depth.

Overall, decision-makers can select any one of the two established definitions, both of which reveal something subtly different about the thermodynamic performance of these systems, while considering the differences between these definitions. The first definition considers the efficiency of the system while paying attention to the variable heat input to the system (by accounting for the varying enthalpy and exergy of the geofluid stream leaving the system). The second definition, on the other hand, evaluates the efficiency of the system while considering only the enthalpy and exergy delivered to the system with the inlet geofluid stream, effectively considering that geofluid stream leaving the system has a ‘lost’ potential.

Figure 12: Comparison of first/second law efficiency definitions for proposed configurations

In closing, Pareto frontiers of optimal solutions for different configurations are shown in Fig. 13, in which the alternative definitions for the thermal and exergy efficiencies are used, in conjunction with the total specific cost and total cost rate, respectively, are considered as objective functions. As can be seen, the overall trends of the optimal solutions for Configurations (b) and (c) are similar to those of the Pareto frontiers of optimal solutions from the perspective of the earlier thermal efficient definitions, shown in Fig. 9. However, the efficiency and cost ranges are different compared to those shown in Fig. 9, and the Pareto optimal solutions have different shapes and trends when different expressions of efficiency are considered. As mentioned earlier, this is important to consider when comparing the various options.

Figure 13: Pareto frontiers of optimal solution points when alternative definitions of thermal and exergy efficiencies are considered as the objective functions

5 Conclusions

In this study, three novel geothermal combined cooling and power (CCP) systems were proposed. The first system (Configuration (a)) comprises an absorption power cycle and an ejector refrigeration cycle, the second (Configuration (b)) a modified Kalina power cycle along with an absorption refrigeration cycle, and the third (Configuration (c)) a double-flash power cycle and ejector refrigeration cycle, in all cases for power generation and cooling, respectively.

The three aforementioned systems were firstly compared comprehensively from both thermodynamic and economic perspectives using parametric and multi-objective particle swarm optimization. The aim of the parametric study was to understand and evaluate the effects of the most influential parameters on the systems' performance. The results revealed that Configuration (c) is particularly sensitive to the evaporator temperature, such that at the high end of investigated temperatures (*i.e.*, 9-10 °C) this system attains a cooling capacity that is even higher than that of Configuration (b), which otherwise has the best cooling performance.

The main findings from the multi-objective optimization study when the overall thermal efficiency and total specific cost are considered as competing objective functions, were:

- Deeper geothermal wells are best suited to optimal Configurations (b) and (c). When considering alternative efficiency definitions relative to the geothermal resource, the optimal depth for Configuration (a) also shifts to deeper wells.
- Overall, the final optimal design of Configuration (a) has the highest power output and exergy efficiency, but the lowest total cost rate and total specific cost. Configuration (b) has the highest thermal efficiency, while having an intermediate cooling capacity. Configuration (c) provides the most cooling, along with reasonable power output, exergy efficiency and total specific cost.
- The final optimal designs of Configurations (a) to (c) selected by the LINMAP method have thermal efficiencies of 19.1%, 43.0%, and 42.4%, respectively, exergy efficiencies of 57.6%, 23.6%, and 33.1%, total cost rate values of 436 \$/h, 558 \$/h, and 596 \$/h, and total specific costs of 29.7 \$/GJ, 66.9 \$/GJ, 43.5 \$/GJ, for well depths of 4.2 km for Configuration (a), and 5.0 km for other two configurations.

- Compared to a conventional geothermal ORC system, Configuration (a) has both higher thermal and exergy efficiencies, a lower total cost rate, and similar power output and specific cost, while also having the potential to deliver limited cooling.

Finally, the main findings when exergy efficiency and total cost are the objective functions, were:

- Shallower geothermal wells are suitable for all configurations ($d = 3.8$ km).
- Overall, the final optimal design of Configuration (a) has the highest power output and exergy efficiency, the lowest total specific cost, and intermediate thermal efficiency and total cost rate, relative to the other systems. Configuration (b) has the highest cooling capacity, and thermal efficiency. Configuration (c) delivers intermediate power and cooling outputs, while having moderate exergy efficiency and total specific cost.
- The final optimal designs of Configurations (a) to (c) selected by the LINMAP method have thermal efficiencies of 13.0%, 29.1%, and 10.5 %, respectively, exergy efficiencies of 67.3%, 30.5%, and 37.3%, total cost rate values of 362 \$/h, 353 \$/h, and 384 \$/h, and total specific costs of 24.9 \$/GJ, 67.5 \$/GJ, 42.7 \$/GJ.
- Compared to conventional geothermal Kalina cycle systems, Configuration (a) has a higher exergy efficiency, but a slightly lower thermal efficiency, and a comparable power output, total cost rate and specific cost; the Kalina system, however, has zero cooling capacity.

In future work, different geothermal temperature gradients can also be considered for different regions which have not been considered in this study. In addition, optimal designs and system performance with integrated component performance modelling [54] as well operational modelling in the presence of heat-source (or sink) intermittency while accounting for changing component performance are important in making predictions close to expectations in practice [55].

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Appendix A

The capital cost of the geothermal system is a function of drilling depth and also the reference power for predicting surface installing, including production and reinjection wells. The drilling cost (\$) is evaluated in this work through a function of depth (m) [56]:

$$Z_{\text{drill}} = (1.72 \times 10^{-7} \times d^2 + 2.3 \times 10^{-3} \times d - 0.62) \times 10^6 \quad (\text{A1})$$

The cost of surface installation (\$) is predicted as a function of the power output (kW) of a conventional ORC-based geothermal system as [57,58]:

$$Z_{\text{SI}} = 2000 \times \dot{W}_{\text{ref}} \times \varphi \quad (\text{A2})$$

The reference power output (kW) is extracted after the curve fitting action considering the temperature of the geothermal well (°C) and mass flow rate (kg/s) [2]:

$$\dot{W}_{\text{ref}} = \dot{m}_{\text{geo}} \times (-67.06 + 0.736458 \times T_{\text{geo,wellhead}}) \quad (\text{A3})$$

It is noted that Eqs. (A2) and (A3) are valid for $d > 3000$ m, and that they should be converted to the present year considering CEPCI for 2010 and the present year.

Furthermore, the total capital cost (as a function of d , and the heat source temperature and flow rate) of the geothermal system is expressed as:

$$Z_{\text{geo}} = Z_{\text{drill}} + Z_{\text{SI}} \quad (\text{A4})$$

The temperature of geothermal wellhead and geothermal brine (heat source temperature in °C) are also predicted considering the measured depth of the geothermal well (d) in meters:

$$T_{\text{geo,wellhead}} = d \times \zeta \quad (\text{A5})$$

$$T_{\text{geo}} = d \times \zeta \times \theta \quad (\text{A6})$$

in which ζ refers to the temperature gradient at the geothermal site (0.038 °C/m in this study) and θ denotes the correction coefficient required to predict the production fluid temperature, which is lower than wellhead temperature [52]. Hence, in Eq. (A6), θ is considered equal to 0.868 to be consistent with the real data by considering the temperature drop across the production well.

The geothermal brine temperature and the costs associated with surface installing and drilling, and overall cost are shown in Fig. A1 considering the borehole geothermal depth as a variable. The

highlighted area is the operating temperature of the proposed medium grade geothermal systems. Producing geothermal brine with a temperature of higher than 200 °C is challenging due to the limitations of drilling and operating and maintenance.

Fig. A1: Geothermal brine temperature and costs associated with the geothermal well

In addition, M_{SP} in the cost function of the separators in Configuration (b) and (c) is expressed as [19]:

$$\log_{10}(M_{SP}) = 3.4974 + 0.4485 \times \log_{10}(V_{SP}[\text{m}^3]) + 0.1074 \times (\log_{10}(V_{SP}[\text{m}^3]))^2 \quad (\text{A7})$$

$$V_{SP}[m^3] = \pi \times (3D_{ip}[m])^2 \times \frac{7 \times D_{ip}[m] + 4.5 \times D_{ip}[m]}{4} \quad (A8)$$

$$D_{ip}[m] = \sqrt{4 \times \frac{A_{ip}[m^2]}{\pi}} \quad (A9)$$

$$A_{ip}[m^2] = \frac{\dot{m}_{in}[kg/s]}{v_t[m/s] \times \rho_{liquid}[kg/m^3]} \quad (A10)$$

$$v_t[m/s] = 0.069 \times \sqrt{\frac{\rho_{liquid}[kg/m^3] - \rho_{vapour}[kg/m^3]}{\rho_{liquid}[kg/m^3]}} \quad (A11)$$

Furthermore, M'_{SP} in the cost function of the separators in Configuration (b) and (c) (see Table A3 and C4), and is given by [19]:

$$M'_{SP} = \max \left\{ \frac{\left(\frac{(P_{in,SP}[\text{bar}] + 1) \times D_{ip}[m]}{2 \times (850 - 0.6 \times (P_{in,SP}[\text{bar}] + 1))} + 0.00315 \right)}{0.0063}, 1 \right\} \quad (A12)$$

The heat exchanger areas required to calculate the capital cost of them is estimated from:

$$\dot{Q} = UA\Delta T_{LMTD} \quad (A13)$$

The value of heat transfer coefficient (U) for different components are listed in Table A1.

Table A1: Values of heat transfer coefficient for different heat exchangers $\left[\frac{\text{kW}}{\text{m}^2 \cdot \text{K}} \right]$

Generator	Heat exchanger	Solution heat exchanger	Absorber	Evaporator	Condenser
1.6	1.0	1.1	0.6	0.9	1.0

Table A2: Cost balance and cost functions for different components of Configuration (a)

Component	Cost balance and auxiliary equations	Cost functions, Z (\$) [59]	Ref. year (CEPCI)
Geothermal section	$\dot{C}_{\text{water}} + \dot{Z}_{\text{geo}} = \dot{C}_1$ $c_{\text{water}} = 0$	See Eqs. (A1) to (A6)	2010 (550.8)
Vapour Generator 1	$\dot{C}_1 + \dot{C}_5 + \dot{Z}_{\text{VG1}} = \dot{C}_6 + \dot{C}_{10} + \dot{C}_2$ $c_1 = c_2$	$Z_{\text{VG1}} = 130 \times (A_{\text{VG1}}[\text{m}^2]/0.093)^{0.78}$	2005 (468.2)
Expansion Valve 2	$\dot{C}_{19} + \dot{Z}_{\text{EV2}} = \dot{C}_{20}$	$Z_{\text{EV2}} = 0$	-
Pump 1	$\dot{C}_{\text{w,pu1}} + \dot{C}_3 + \dot{Z}_{\text{pu1}} = \dot{C}_4$ $c_{\text{w,pu1}} = c_{\text{w,tu}}$	$Z_{\text{pu1}} = 3540 \times (\dot{W}_{\text{pu1}}[\text{kW}])^{0.71}$	2010 (550.8)
Pump 2	$\dot{C}_{\text{w,pu2}} + \dot{C}_{14} + \dot{Z}_{\text{pu2}} = \dot{C}_{15}$ $c_{\text{w,pu2}} = c_{\text{w,tu}}$	$Z_{\text{pu2}} = 3540 \times (\dot{W}_{\text{pu2}}[\text{kW}])^{0.71}$	2010 (550.8)
Absorber	$\dot{C}_9 + \dot{C}_{11} + \dot{C}_{12} + \dot{Z}_{\text{abs}} = \dot{C}_{13} + \dot{C}_3$ $\frac{\dot{C}_{11} + \dot{C}_9}{\dot{E}_{11} + \dot{E}_9} = \frac{\dot{C}_3}{\dot{E}_3}, c_{12} = 0$	$Z_{\text{abs}} = 130 \times (A_{\text{abs}}[\text{m}^2]/0.093)^{0.78}$	2005 (468.2)
Expansion Valve 1	$\dot{C}_8 + \dot{Z}_{\text{EV1}} = \dot{C}_9$	$Z_{\text{EV1}} = 0$	-
Turbine	$\dot{C}_{10} + \dot{Z}_{\text{tu}} = \dot{C}_{11} + \dot{C}_{\text{w,tu}}$ $c_{10} = c_{11}$	$Z_{\text{tu}} = 4405 \times (\dot{W}_{\text{tu}}[\text{kW}])^{0.7}$	2003 (402.3)
Solution Heat Exchanger	$\dot{C}_4 + \dot{C}_7 + \dot{Z}_{\text{SHE}} = \dot{C}_8 + \dot{C}_5$ $c_8 = c_7$	$Z_{\text{SHE}} = 130 \times (A_{\text{SHE}}[\text{m}^2]/0.093)^{0.78}$	2005 (468.2)
Condenser	$\dot{C}_{17} + \dot{C}_{25} + \dot{Z}_{\text{con}} = \dot{C}_{26} + \dot{C}_{18}$ $c_{17} = c_{18}, c_{25} = 0$	$Z_{\text{con}} = 130 \times (A_{\text{con}}[\text{m}^2]/0.093)^{0.78}$	2005 (468.2)
Evaporator	$\dot{C}_{20} + \dot{C}_{21} + \dot{Z}_{\text{eva}} = \dot{C}_{22} + \dot{C}_{23}$ $c_{20} = c_{23}, c_{21} = 0$	$Z_{\text{eva}} = 130 \times (A_{\text{eva}}[\text{m}^2]/0.093)^{0.78}$	2005 (468.2)
Ejector	$\dot{C}_{16} + \dot{C}_{24} + \dot{Z}_{\text{eje}} = \dot{C}_{17}$	$Z_{\text{EV2}} = 0$	-
Booster	$\dot{C}_{\text{w,b}} + \dot{C}_{23} + \dot{Z}_{\text{b}} = \dot{C}_{24}$ $c_{\text{w,b}} = c_{\text{w,tu}}$	$Z_{\text{b}} = 9624.2 \times (\dot{W}_{\text{b}}[\text{kW}])^{0.46}$	2005 (468.2)
Vapour Generator 2	$\dot{C}_6 + \dot{C}_{15} + \dot{Z}_{\text{VG2}} = \dot{C}_7 + \dot{C}_{16}$ $c_7 = c_6$	$Z_{\text{VG2}} = 130 \times (A_{\text{VG2}}[\text{m}^2]/0.093)^{0.78}$	2005 (468.2)

Table A3: Cost balance and cost functions for different components of Configuration (b)

Component	Cost balance and auxiliary equations	Cost functions, Z (\$) [59]	Ref. year (CEPCI)
Geothermal section	$\dot{C}_{\text{water}} + \dot{Z}_{\text{geo}} = \dot{C}_1$ $c_{\text{water}} = 0$	See Eqs. (A1) to (A6)	2010 (550.8)
Vapour Generator	$\dot{C}_6 + \dot{C}_2 + \dot{Z}_{\text{VG}} = \dot{C}_3 + \dot{C}_{17} + \dot{C}_7$ $c_2 = c_3$	$Z_{\text{VG}} = 130 \times (A_{\text{VG}}[\text{m}^2]/0.093)^{0.78}$	2005 (468.2)
Expansion Valve 2	$\dot{C}_{18} + \dot{Z}_{\text{EV2}} = \dot{C}_{19}$	$Z_{\text{EV2}} = 0$	—
Pump 1	$\dot{C}_{\text{w,pu1}} + \dot{C}_4 + \dot{Z}_{\text{pu1}} = \dot{C}_5$ $c_{\text{w,pu1}} = c_{\text{w,tu}}$	$Z_{\text{pu1}} = 3540 \times (\dot{W}_{\text{pu1}}[\text{kW}])^{0.71}$	2010 (550.8)
Pump 2	$\dot{C}_{\text{w,pu2}} + \dot{C}_{15} + \dot{Z}_{\text{pu2}} = \dot{C}_{16}$ $c_{\text{w,pu2}} = c_{\text{w,tu}}$	$Z_{\text{pu2}} = 3540 \times (\dot{W}_{\text{pu2}}[\text{kW}])^{0.71}$	2010 (550.8)
Absorber	$\dot{C}_{16} + \dot{C}_{20} + \dot{C}_{27} + \dot{Z}_{\text{abs}} = \dot{C}_4 + \dot{C}_{28}$ $\frac{\dot{C}_{16} + \dot{C}_{20}}{\dot{E}_{16} + \dot{E}_{20}} = \frac{\dot{C}_4}{\dot{E}_4}, c_{27} = 0$	$Z_{\text{abs}} = 130 \times (A_{\text{abs}}[\text{m}^2]/0.093)^{0.78}$	2005 (468.2)
Expansion Valve 1	$\dot{C}_{12} + \dot{Z}_{\text{EV1}} = \dot{C}_{13}$	$Z_{\text{EV1}} = 0$	—
Turbine	$\dot{C}_9 + \dot{Z}_{\text{tu}} = \dot{C}_{10} + \dot{C}_{\text{w,tu}}$ $c_9 = c_{10}$	$Z_{\text{tu}} = 4405 \times (\dot{W}_{\text{tu}}[\text{kW}])^{0.7}$	2003 (402.3)
Solution Heat Exchanger	$\dot{C}_{11} + \dot{C}_5 + \dot{Z}_{\text{SHE}} = \dot{C}_6 + \dot{C}_{12}$ $c_{12} = c_{11}$	$Z_{\text{SHE}} = 130 \times (A_{\text{SHE}}[\text{m}^2]/0.093)^{0.78}$	2005 (468.2)
Condenser 1	$\dot{C}_{17} + \dot{C}_{23} + \dot{Z}_{\text{con1}} = \dot{C}_{24} + \dot{C}_{18}$ $c_{17} = c_{18}, c_{23} = 0$	$Z_{\text{con1}} = 130 \times (A_{\text{con1}}[\text{m}^2]/0.093)^{0.78}$	2005 (468.2)
Condenser 2	$\dot{C}_{14} + \dot{C}_{25} + \dot{Z}_{\text{con2}} = \dot{C}_{26} + \dot{C}_{15}$ $c_{14} = c_{15}, c_{25} = 0$	$Z_{\text{con2}} = 130 \times (A_{\text{con}}[\text{m}^2]/0.093)^{0.78}$	2005 (468.2)
Evaporator	$\dot{C}_{19} + \dot{C}_{21} + \dot{Z}_{\text{eva}} = \dot{C}_{22} + \dot{C}_{20}$ $c_{20} = c_{19}, c_{21} = 0$	$Z_{\text{eva}} = 130 \times (A_{\text{eva}}[\text{m}^2]/0.093)^{0.78}$	2005 (468.2)
Mixer	$\dot{C}_{14} = \dot{C}_{13} + \dot{C}_{10} + \dot{Z}_{\text{mix}}$	$Z_{\text{mix}} = 0$	—
Heat Exchanger	$\dot{C}_1 + \dot{C}_7 + \dot{Z}_{\text{HE}} = \dot{C}_2 + \dot{C}_8$ $c_1 = c_2$	$Z_{\text{HE}} = 130 \times (A_{\text{SHE}}[\text{m}^2]/0.093)^{0.78}$	2005 (468.2)
Separator	$\dot{C}_8 + \dot{Z}_{\text{SP}} = \dot{C}_9 + \dot{C}_{11}$	$Z_{\text{SP}} = (2.52 + 1.82 \times M'_{\text{SP}}) \times M_{\text{SP}}$	2000 (394.1)

Table A4: Cost balance and cost functions for different components of Configuration (c)

Component	Cost balance and auxiliary equations	Cost functions, Z (\$) [59]	Ref. year (CEPCI)
Geothermal section	$\dot{C}_{\text{water}} + \dot{Z}_{\text{geo}} = \dot{C}_1$ $c_{\text{water}} = 0$	See Eqs. (A1) to (A6)	2010 (550.8)
Separator 1	$\dot{C}_2 + \dot{Z}_{\text{SP1}} = \dot{C}_3 + \dot{C}_4$ $c_3 = c_4$	$Z_{\text{SP1}} = (2.52 + 1.82 \times M'_{\text{SP1}}) \times M_{\text{SP1}}$	2000 (394.1)
Separator 2	$\dot{C}_5 + \dot{Z}_{\text{SP2}} = \dot{C}_6 + \dot{C}_9$ $c_6 = c_9$	$Z_{\text{SP2}} = (2.52 + 1.82 \times M'_{\text{SP2}}) \times M_{\text{SP2}}$	2000 (394.1)
Expansion Valve 2	$\dot{C}_4 + \dot{Z}_{\text{EV2}} = \dot{C}_5$	$Z_{\text{EV2}} = 0$	—
Expansion Valve 1	$\dot{C}_1 + \dot{Z}_{\text{EV1}} = \dot{C}_2$	$Z_{\text{EV1}} = 0$	—
Pump	$\dot{C}_{\text{w,pu}} + \dot{C}_{16} + \dot{Z}_{\text{pu}} = \dot{C}_{17}$ $c_{\text{w,pu}} = c_{\text{w,tu}}$	$Z_{\text{pu}} = 3540 \times (\dot{W}_{\text{pu}}[\text{kW}])^{0.71}$	2010 (550.8)
Turbine	$\dot{C}_3 + \dot{C}_6 + \dot{Z}_{\text{tu}} = \dot{C}_7 + \dot{C}_{\text{w,tu}}$ $\frac{\dot{C}_6 + \dot{C}_3}{\dot{E}_6 + \dot{E}_3} = \frac{\dot{C}_7}{\dot{E}_7}$	$Z_{\text{tu}} = 4405 \times (\dot{W}_{\text{tu}}[\text{kW}])^{0.7}$	2003 (402.3)
Condenser 1	$\dot{C}_7 + \dot{C}_{19} + \dot{Z}_{\text{con1}} = \dot{C}_8 + \dot{C}_{20}$ $c_7 = c_8, c_{19} = 0$	$Z_{\text{con1}} = 130 \times (A_{\text{con1}}[\text{m}^2]/0.093)^{0.78}$	2005 (468.2)
Valve 1	$\dot{C}_8 + \dot{C}_{17} + \dot{Z}_{\text{va1}} = \dot{C}_{18}$	$Z_{\text{va1}} = 0$	—
Valve 2	$\dot{C}_{12} + \dot{C}_{16} + \dot{Z}_{\text{va2}} = \dot{C}_{11}$	$Z_{\text{va2}} = 0$	—
Cooling set	$\dot{C}_9 + \dot{C}_{21} + \dot{Z}_{\text{CS}} + \dot{C}_{23} = \dot{C}_{22} + \dot{C}_{24} + \dot{C}_{16}$ $c_9 = c_{16}, c_{24} = c_{22}, c_{\text{w,b}} = c_{\text{w,tu}}$	$\dot{Z}_{\text{CS}} = \dot{Z}_{\text{eva}} + \dot{Z}_{\text{eje}} + \dot{Z}_{\text{con2}} + \dot{Z}_{\text{EV3}} + \dot{Z}_{\text{com}}$ $Z_{\text{com}} = 91562 \times (\dot{W}_{\text{com}}[\text{kW}]/455)^{0.67}$	Refer to each component