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ARTICLE



Performance analysis and sensitivity of system parameters on the performance of trilateral-cycle power generator systems

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ABSTRACT

Though the trilateral cycle (TLC) is a promising heat recovery-to-power cycle, its application has not been widely accepted or commercialised due to some thermodynamic feasibility concerns. This study examined the performance analysis and sensitivity of the system parameters on the thermodynamic performance of the TLC power generator systems for waste heat recovery-to-power generation. Thermodynamic models of the simple, recuperative, reheat and regenerative TLCs were established. Performance analysis and sensitivity of the system parameters on the cycles' performance were conducted at the expander inlet temperature of 453 to 473 K, expander pressure of 2 to 3 MPa and expander isentropic efficiency of 50% to 100%. The expander inlet temperatures, pressures, and its isentropic efficiencies have significant effects on the thermodynamic efficiencies and net work output of the cycles. At 473 K cycle high temperature, the thermal efficiencies of the cycles increase from 20.13% to 21.97%, 23.29% to 23.91%, 20.62% to 22.07% and 20.66% to 22.9% for the simple, recuperative, reheat and regenerative TLCs, respectively. Their corresponding net power outputs varied from 131.6 to 134.1 kW, 145.9 to 152.2 kW, 113.8 to 124.1 kW and 124.9 to 130.5 kW, respectively. The cycles' thermodynamic performance increased with an increase in expander inlet temperature, expander pressure and expander isentropic efficiencies.

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Trilateral cycle; waste heat recovery; heat recovery-to-power generation; performance metrics; sensitivity analysis

1. Introduction

The mounting pressures and necessity for innovative energy conversion technologies and the need to bridge the rising gap between energy demand and supply and its affordable and security of supply have renewed the growing interest in low-grade waste heat recovery-to-power technologies (Ajimotokan 2014, 2016; Ajimotokan and Sher 2015). This is because waste heat recovery-to-power technologies are promising alternatives for enhancing system efficiency (Ajimotokan 2014, 2016) and reducing the unit cost of energy (Ajimotokan 2014; Demirkaya et al. 2011), and consequently mitigating the severe environmental issues due to reduction in greenhouse gases emission (Ajimotokan 2014, 2016; Fang et al. 2013; Oloyede and Ajimotokan 2016; Patil, Ajah, and Herder 2009; Rahbar et al. 2017) and power generation carbon footprint (Ziviani, Beyene, and Venturini 2014).

Moreover, there are abundant sources of alternative energy such as waste heat dissipated from about 20% to 50% of the total used primary energy or the heat produced during industrial processes (Fang et al. 2013; Johnson, Choate, and Dillich 2008; Lian, Li, and Shu 2011) and renewable energy such as geothermal and solar thermal (Ajao et al. 2009; Ajimotokan 2014; Fischer 2011; Omosewo, Olaoye, and Ajibola 2017; Song, Gu, and Ren 2016; Yari 2009). These numerous

heat energy sources, which are generally low-grade heat, are significant in quantities (Ajimotokan 2014; Chen 2010), and they are underutilised due to the non-existence of a resourcefully efficient waste heat recovery-to-power generator system (Chen et al. 2011; Franco and Casarosa 2002). Thus, a number of innovative solutions such as the organic Rankine cycle, supercritical Rankine cycle and trilateral cycle have been proposed and/or developed to enhance system (overall) efficiency, and to gain on a smaller scale, comparable advantages of efficiency (Ajimotokan 2014; Ajimotokan and Sher 2015; Franco and Casarosa 2002), where the performance of the conventional Rankine-cycle power generator systems is generally inefficient (Franco and Casarosa 2002).

Nonetheless, the necessity for a power cycle to attain optimum reversible work from a non-isothermal single-phase heat source has led to the conception of the trilateral cycle (TLC), an advanced (modified) Rankine cycle for waste heat recovery-to-power applications (Ajimotokan 2014; Smith, Stosic, and Kovacevic 2005). Although, since the 19th century, the proposition of heat recovery-to-power cycle as an open cycle has been figured out (Smith, Stosic, and Kovacevic 2005); Ajimotokan (2014) and Smith (1993) among others have described the TLC as a promising alternative for waste heat recovery-to-power

generation, able to achieve this ideal. This cycle, often called the trilateral wet vapour, or trilateral flash cycle is a power cycle, proposed to improve the heat recovery-to-power performance, exceptionally from non-isothermal heat sources (Ajimotokan 2014), which is beneficial for decentralised applications, in particular, where the power-to-weight ratio of the power plant is of considerable importance. Typically, the TLC has similar key components as the Rankine-cycle power generator systems. Unlike the Rankine cycles, it neither uses a boiler nor evaporates the working fluid during the heating phase, but flash expands it as a two-phase mixture at the saturated liquid state (Ajimotokan 2014, 2016; Ajimotokan and Sher 2015; Zamfirescu and Dincer 2008). The bypass of isothermal boiling phase in the TLC during operation allows a heat transfer with no pinch-point limitation, which causes thermal march among the working fluid and the heat source exergy temperature profiles (Ajimotokan and Sher 2015; Zamfirescu and Dincer 2008), minimises exergy destruction.

The application of the TLC has not been widely accepted or commercialised due to thermodynamic and technical feasibilities of expanding a two-phase mixture as well as safety concerns. The most technically challenging TLC component – the expander can be a scroll or screw expander, a variable phase turbine or reciprocating engine (Fischer 2011). The efficient isentropic expansion of light hydrocarbons such as *n*-pentane into the two-phase region from the saturated liquid state determines the feasibility of the cycle, which can be accomplished efficiently employing Lysholm twin-screw expander, in particular, whenever the exhaust vapour is wet (Smith and da Silva 1994). Also, the screw expander has been studied and proposed as an appropriate solution for the application in TLC power generator systems because it can attain over 75% isentropic efficiency (Smith, Stosic, and Kovacevic 2001). Likewise, expanders employed in LNG applications comprising two-phase fluids expansion accomplish about 85% isentropic efficiency, which are available from original equipment manufacturers (Ajimotokan, Sher, and Biliyok 2014; Kaupert, Gandhi, and Kaehler 2013).

Though a review of relevant theory and literature underscores the significant studies that have been carried out in the research and development of the TLC technology. It also highlighted the substantial progress accomplished in the recent past, primarily in the areas of possible applications of the TLC technology (Ajimotokan, Sher, and Biliyok 2014; Smith, Stosic, and Kovacevic 2005), its two-phase expansion machines modelling and design (Li et al. 2012), process modelling, simulation and optimisation of the TLCs (Ajimotokan and Sher 2015), development of TLC (Bianchi, McGinty, and Oliver 2017; Iqbal et al. 2017; Yu et al. 2018), process development and

thermal integration of simple TLC (Ajimotokan 2014; Ajimotokan and Sher 2015), and thermodynamic (efficiency) analysis (Ajimotokan 2016). Notwithstanding, there are gaps in knowledge on the effects of system parameters and their sensitivity on performance metrics of the TLC power generator systems.

Thus, this paper presents the performance analysis and sensitivity of system parameters on performance metrics of the TLC power generator systems for low-grade waste heat recovery-to-power generation. This is due to the significance of the effects of system parameters on the TLC power generator systems, which often vary with the thermodynamic performance metrics. The thermodynamic efficiencies of the TLC power generator systems and their net work outputs and the sensitivity analysis of system parameters of the TLCs were investigated.

2. Methodology

The trilateral-cycle (TLC) power generator systems for low-grade waste heat recovery-to-power generation were studied. Thermodynamic models of four different TLC configurations, including the simple, recuperative, reheat and regenerative TLC power generator systems were developed and implemented using EES (engineering equation solver). The performance analysis of these TLC power generator systems and the sensitivity analysis of their system parameters at an expander inlet temperature of 453 to 473 K, the pressure of 2 to 3 MPa and customised isentropic efficiency of 50% to 100% employing *n*-pentane were carried out and analysed.

2.1. System description

Figure 1 depicts the TLCs schematic configuration and their temperature–entropy (*T*–*s*) diagrams. Figure 1 (a) depicts the simple TLC schematic configuration and its *T*–*s* diagram. The simple TLC's key components, similar to the Rankine cycles, include the feed pump, heater, two-phase expander and condenser. It has four thermodynamic processes, comprising working fluid isentropic compression in the feed pump (process 1 to 2); that is, the transfer of energy from the feed pump to the working fluid; high-temperature heat addition at constant pressure (process 2 to 3); isentropic expander expansion of compressed and heated working fluid (process 3 to 4); that is, the transfer of energy from the working fluid to the rotor work of the expander; and working fluid low-temperature heat rejection at constant pressure in the condenser (process 4 to 1). Figure 1(b) depicts the recuperative TLC schematic configuration and its *T*–*s* diagram. The key components of the recuperative TLC include feed pump, recuperator, heater, two-

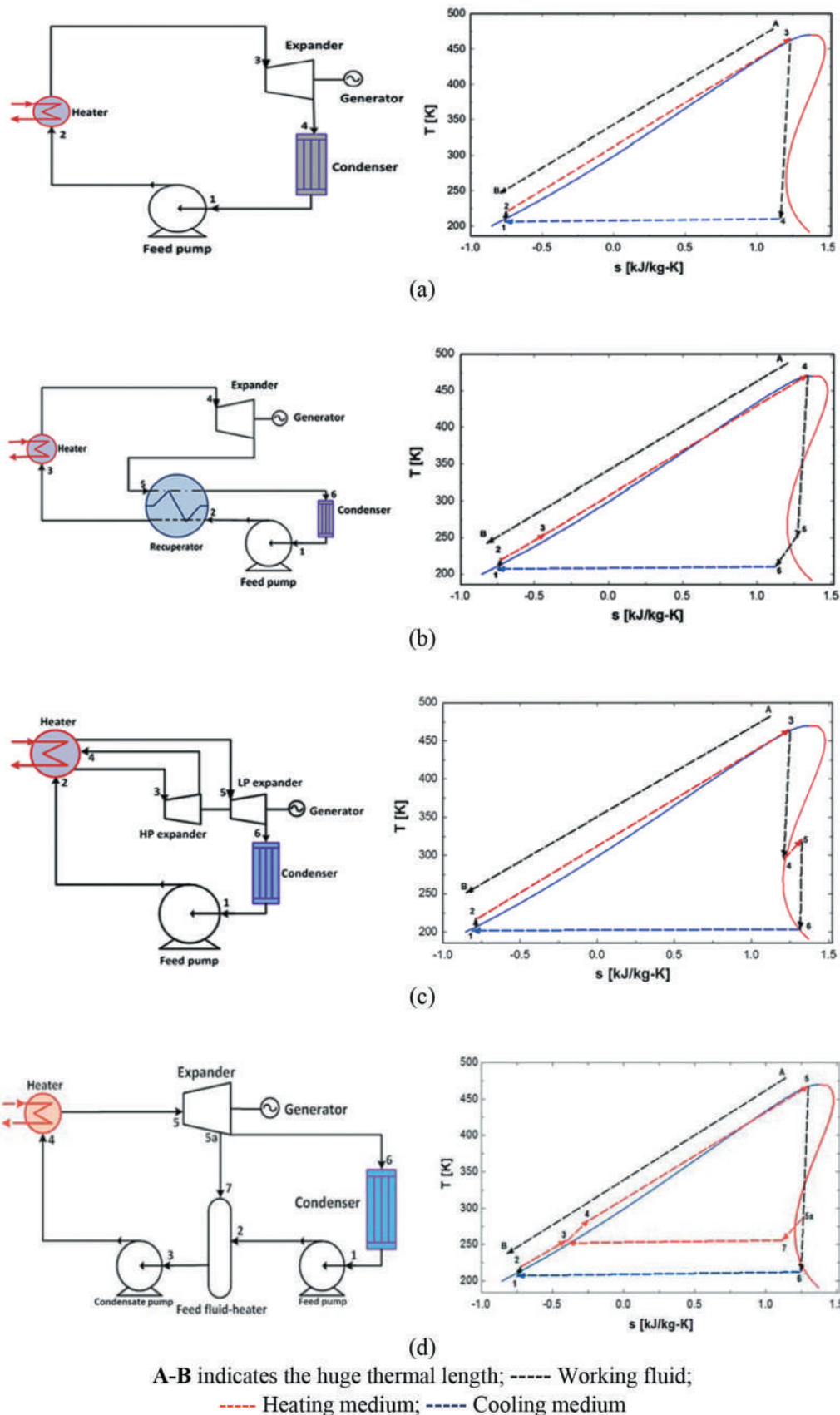


Figure 1. Schematic cycle configurations and their T-s diagrams for (a) simple TLC; (b) recuperative TLC; (c) reheat TLC; and (d) regenerative TLC.

phase expander and condenser. It has five thermodynamic processes, comprising the addition of a recuperator (internal heat exchanger) to the simple TLC, which is the expander exhaust latent heat recuperation

at constant pressure by the recuperator; that is, states 2 and 5. Figure 1(c) depicts the reheat TLC schematic configuration and its T-s diagram. The key components of the reheat TLC include feed pump, heater,

two-phase high- and low-pressure expanders, and condenser. It has five thermodynamic processes, comprising the high-pressure expander isentropic expansion of compressed and heated working fluid (process 3 to 4), and low-pressure expander isentropic expansion of compressed and reheated working fluid (process 4 to 5) to the simple TLC. Figure 1(d) depicts the regenerative TLC schematic configuration and its T-s diagram. The key components of the regenerative TLC include feed pump, feed-fluid heater, condensate pump, heater, two-phase expander, and condenser. It has six thermodynamic processes, comprising the addition of feed-fluid heater and condensate pump to the simple TLC, which are the working fluid isentropic compression in the condensate pump (process 3 to 4) and low-temperature fluid bleeding at constant pressure for preheating of the compressed liquid at the feed pump outlet – a regeneration process; that is, states 5a to 7.

2.2. System modelling

The TLCs' key components, corresponding to their thermodynamic processes (see Figure 1), were modelled employing EES (Ajimotokan 2014; Ajimotokan and Sher 2015). Based on experimental rule-of-thumb and thermodynamic requirements subject to the engineering constraint imposed by the TLCs, the following specific philosophical assumptions were made:

- (i) The TLCs and their components operate at steady-state conditions;
- (ii) There is a thermodynamic equilibrium at the inlets and outlets of each component;
- (iii) The kinetic and potential energy variations of the heat transfer and working fluid in the cycles are negligible; and
- (iv) The systems' heat loss and pressure drop are negligible (in the heat exchanger and connecting pipes).

The models of the simple TLC can be described as follows: The specific work output by the feed pump $\dot{W}_{fp}$; that is, the thermodynamic process 1 to 2 of the working fluid in the feed pump can be expressed using Eq. (1) as follows:

$$\dot{W}_{fp} = \frac{\dot{m}_{wf}(h_{fp, is}^{out} - h_{fp}^{in})}{\eta_{fp}} \quad (1)$$

The heat transfer rate by the heat exchanger $\dot{Q}_{in}$; that is, the thermodynamic process 2 to 3 in the heat exchanger can be expressed using Eq. (2) as follows:

$$\dot{Q}_{in} = \dot{m}_{wf}(h_{hx}^{out} - h_{hx}^{in}) \quad (2)$$

The specific work output by the expander $\dot{W}_{exp}$; that is, the thermodynamic process 3 to 4 in the two-phase organic fluid expander can be expressed using Eq. (3) as follows:

$$\dot{W}_{exp} = \dot{m}_{wf}(h_{exp}^{in} - h_{exp, is}^{out})\eta_{exp} \quad (3)$$

The heat transfer rate by the condenser $\dot{Q}_{out}$; that is, the thermodynamic process 4 to 1 in the condenser can be expressed using Eq. (4) as follows:

$$\dot{Q}_{out} = \dot{m}_{wf}(h_{con}^{in} - h_{con}^{out}) \quad (4)$$

where $\dot{m}_{wf}$ is the mass flow rate of the working fluid, h_{fp}^{in} is the feed pump-specific enthalpy at its inlet, $h_{fp, is}^{out}$ is the specific enthalpy of the feed pump's isentropic efficiency at its outlet, h_{hx}^{in} and h_{hx}^{out} are the heat exchanger-specific enthalpies at its inlet and outlet, respectively, h_{exp}^{in} is the expander-specific enthalpy at its inlet, $h_{exp, is}^{out}$ is the specific enthalpy of the expander's isentropic efficiency at its outlet, h_{con}^{in} and h_{con}^{out} are the condenser-specific enthalpies at the inlet and outlet, respectively, η_{fp} is the feed pump isentropic efficiency and η_{exp} is the expander isentropic efficiency.

The models of the recuperative TLC, which is the same simple TLC with the recuperator integration can be described as follows. Thus, the heat transfer rate by the recuperator $\dot{Q}_{recup}$; that is, the thermodynamic process 5 to 6 in the heat exchanger can be expressed using Eq. (5) as follows:

$$\dot{Q}_{recup} = \dot{m}_{wf}(h_{recup}^{out} - h_{recup}^{in}) \quad (5)$$

The models of the reheating TLC, which is the same simple TLC except the integration of low- and high-pressure expanders and reheating process can be described as follows. Thus, the specific work output by the high-pressure expander $\dot{W}_{exp, hp}$; that is, the thermodynamic process 3 to 4 in the two-phase organic fluid high-pressure expander can be expressed using Eq. (6) as follows:

$$\dot{W}_{exp, hp} = \dot{m}_{wf}(h_{exp, hp}^{in} - h_{exp, hp, is}^{out})\eta_{exp, hp} \quad (6)$$

The rate of heat transfer by the heat exchanger $\dot{Q}_{rh}$ during reheating; that is, the thermodynamic process 4 to 5 of the reheating in the heat exchanger can be expressed using Eq. (7) as follows:

$$\dot{Q}_{rh} = \dot{m}_{wf}(h_{hx}^{out} - h_{hx}^{in}) \quad (7)$$

The specific work output by the low-pressure expander $\dot{W}_{exp, lp}$; that is, the thermodynamic process 5 to 6 in the two-phase organic fluid low-pressure expander can be expressed using Eq. (8) as follows:

$$\dot{W}_{exp, lp} = \dot{m}_{wf}(h_{exp, lp}^{in} - h_{exp, lp, is}^{out})\eta_{exp, lp} \quad (8)$$

The models of the regenerative TLC, which is the same simple TLC except the integration of a feed-fluid heater and condensate pump can be described as follows. Thus, the rate of heat transfer by the feed-fluid heat exchanger $\dot{Q}_{fhx}$; that is, the thermodynamic process 2 to 3 in the feed-fluid heat exchanger can be expressed using Eq. (9) as follows:

$$\dot{Q}_{fhx} = \dot{m}_{wf} (h_{fhx}^{out} - h_{fhx}^{in}) \quad (9)$$

The specific work output by the condensate pump $\dot{W}_{cp}$; that is, the thermodynamic process 3 to 4 of the working fluid in the condensate pump can be expressed using Eq. (10) as follows:

$$\dot{W}_{cp} = \frac{\dot{m}_{wf} (h_{cp-is}^{out} - h_{cp}^{in})}{\eta_{cp}} \quad (10)$$

At the subcritical operating conditions, the model simulation with input parameters was implemented and results obtained. The net work output $\dot{W}_{net}$ of the simple and recuperative TLCs can be expressed using Eq. (11) as follows:

$$\dot{W}_{net} = \dot{W}_{exp} - \dot{W}_{fp} \quad (11)$$

The net work output $\dot{W}_{net}$ of the reheat TLC can be expressed using Eq. (12) as follows:

$$\dot{W}_{net} = \dot{W}_{exp,hp} + \dot{W}_{exp,lp} - \dot{W}_{fp} \quad (12)$$

The net work output $\dot{W}_{net}$ of the regenerative TLC can be expressed using Eq. (13) as follows:

$$\dot{W}_{net} = \dot{W}_{exp} - (\dot{W}_{fp} + \dot{W}_{cp}) \quad (13)$$

The thermal efficiency η_{th} of the simple and recuperative TLCs can be expressed using Eq. (14) as follows:

$$\eta_{th} = \frac{\dot{W}_{exp} - \dot{W}_{fp}}{\dot{Q}_{in}} \quad (14)$$

The thermal efficiency η_{th} of the reheat TLC can be expressed using Eq. (15) as follows:

$$\eta_{th} = \frac{\dot{W}_{exp,hp} + \dot{W}_{exp,lp} - \dot{W}_{fp}}{\dot{Q}_{hx} + \dot{Q}_{rh}} \quad (15)$$

The thermal efficiency η_{th} of the regenerative TLC can be expressed using Eq. (16) as follows:

$$\eta_{th} = \frac{\dot{W}_{exp} - (\dot{W}_{fp} + \dot{W}_{cp})}{\dot{Q}_{fhx} + \dot{Q}_{hx}} \quad (16)$$

The exergy efficiency η_{II} of the cycles can be expressed using Eq. (17) as follows:

$$\eta_{II} = \frac{\eta_{th}}{\eta_{th,rev}} \quad (17)$$

and the reversible thermal efficiency $\eta_{th,rev}$ (Carnot equivalent) can be expressed using Eq. (18) as follows:

$$\eta_{th,rev} = \left(1 - \frac{T_{hsk}}{T_{hs}} \right) \quad (18)$$

where T_{hsk} and T_{hs} is the heat (cold stream) sink temperature and heat source temperature.

2.3. Performance analysis and sensitivity of system parameters

The standard thermodynamic performance simulations, at the subcritical operating conditions, were implemented with the defined model input parameters. Afterwards, an exhaustive performance analysis of the cycles was conducted with the key models' input parameters: the expander inlet temperature, expander inlet pressure and expander isentropic efficiency to evaluate the effects of these parameters on the cycles' performance metrics. The sensitivity of the system parameters on performance metrics of TLC power generator systems was conducted at an expander inlet temperature of 453 to 473 K, the expander inlet pressure of 2 to 3 MPa and expander isentropic efficiency of 50% to 100% employing *n*-pentane as the working fluid. These limits were based on the applicability range of the working fluid within its thermo-physical properties and the chemical stability within a desirable range of temperature. The cycle high-temperature upper limit depends on the fluid thermal stability and its lower limit on the thermo-economic factors such as the size of the heat exchangers required for the heat addition to and rejection from the working fluid.

Using the power cycle's analysis and the thermo-physical properties of the working fluid, the sensitivity of the subcritical operating system parameters was determined for total heat transfer capacity; i.e. thermodynamic efficiencies, and expander specific power factor; i.e. cycle net work output per unit mass flow rate of working fluid. The sensitivity of the system parameters on performance metrics of TLC power generator systems was carried out using the standard thermodynamic performance simulations. These were implemented by varying the expander inlet pressure over 2 to 3 MPa at the cycle high temperatures of 463 K, 468 K and 473 K, respectively. Figure 2 depicts the flow chart of the simulation algorithm for the performance analysis and parametric sensitivity studies of the TLC power generator systems.

3. Results and discussion

From the thermodynamic parametric simulation experimentations, the behavioural curves of the performance metrics of the trilateral-cycle (TLC) power generator systems were presented to depict the effects of varying expander inlet pressures at the cycle high temperature and expander isentropic efficiency on the

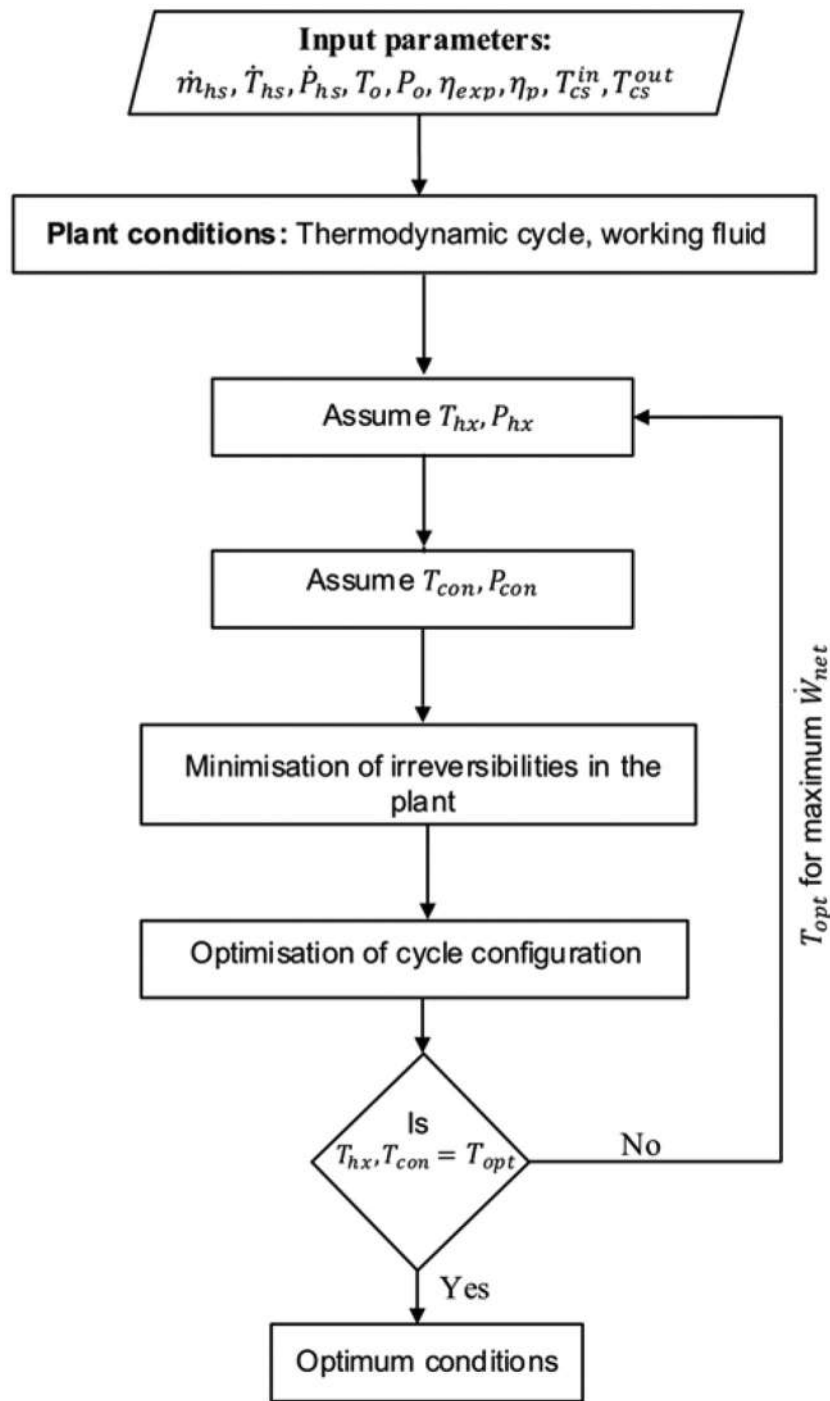


Figure 2. The flow chart of the simulation algorithm for performance analysis and parametric sensitivity studies.

system performance. The parametric sensitivity studies were performed by varying the expander inlet pressure over 2 to 3 MPa at the cycle high temperatures of 463 K, 468 K and 473 K, respectively.

Figure 3 depicts the performance metrics of the different TLCs. Figure 3(a) depicts that the simple, recuperative, reheat and regenerative TLCs achieved a thermal efficiency of 21.97%, 23.91%, 22.07%, and 22.9%, respectively. The recuperative TLC exhibited the highest thermal efficiency (23.91%) while the simple TLC exhibited the lowest (21.97%). Figure 3(b) depicts that the simple, recuperative, reheat and regenerative TLCs attained an exergy efficiency of 63.34%,

68.96%, 63.65% and 66.05%, respectively. The recuperative TLC exhibited the highest exergy efficiency (68.96%) while the simple TLC exhibited the lowest (63.34%). Figure 3(c) depicts that the simple, recuperative, reheat and regenerative TLCs attained a net work output of 134.1 kW, 152.2 kW, 124.1 kW, and 130.5 kW, respectively. The recuperative TLC exhibited the highest net work output (152.2 kW) while the reheat TLC exhibited the lowest (124.1 kW). A comparison among these cycles depicted that the recuperative TLC exhibited the best performance metrics, followed by regenerative TLC due to the lower irreversibility rates in their heat exchangers and the

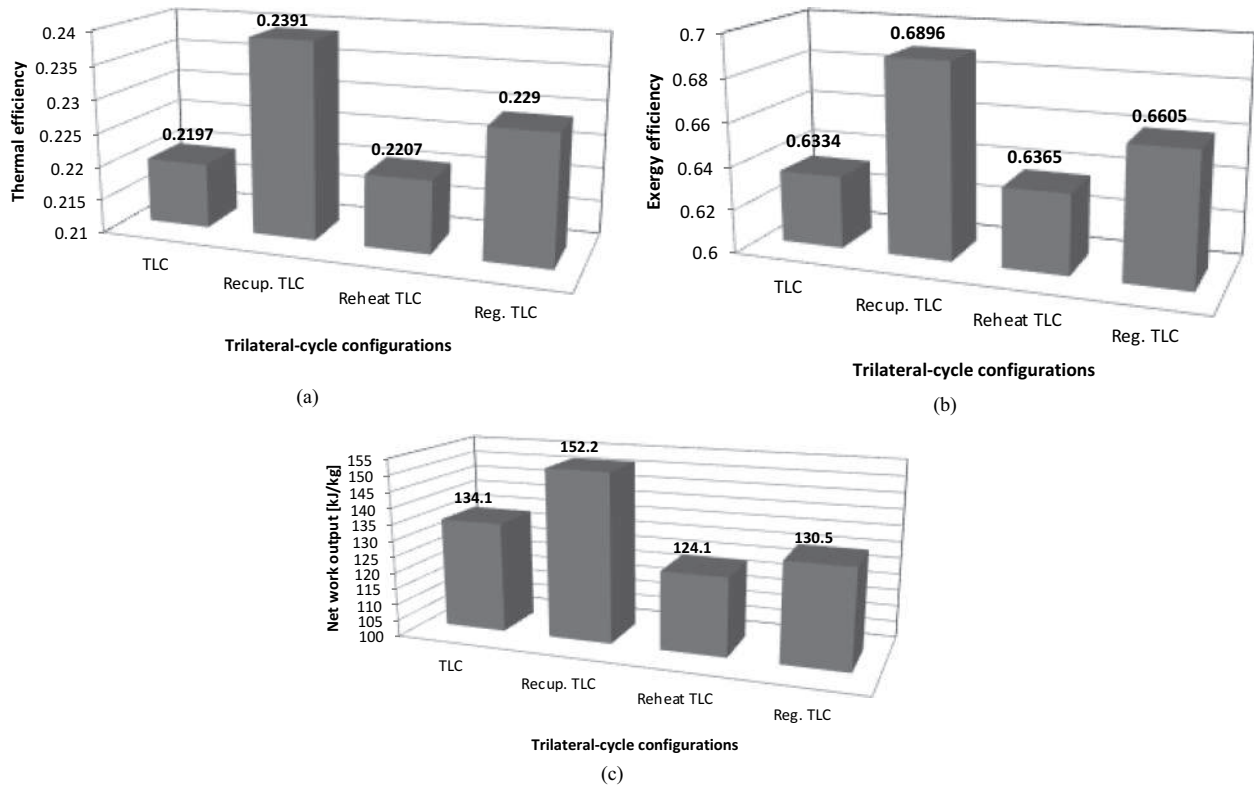


Figure 3. The performance metrics of the TLCs for (a) thermal efficiency; (b) exergy efficiency; and (c) net work output.

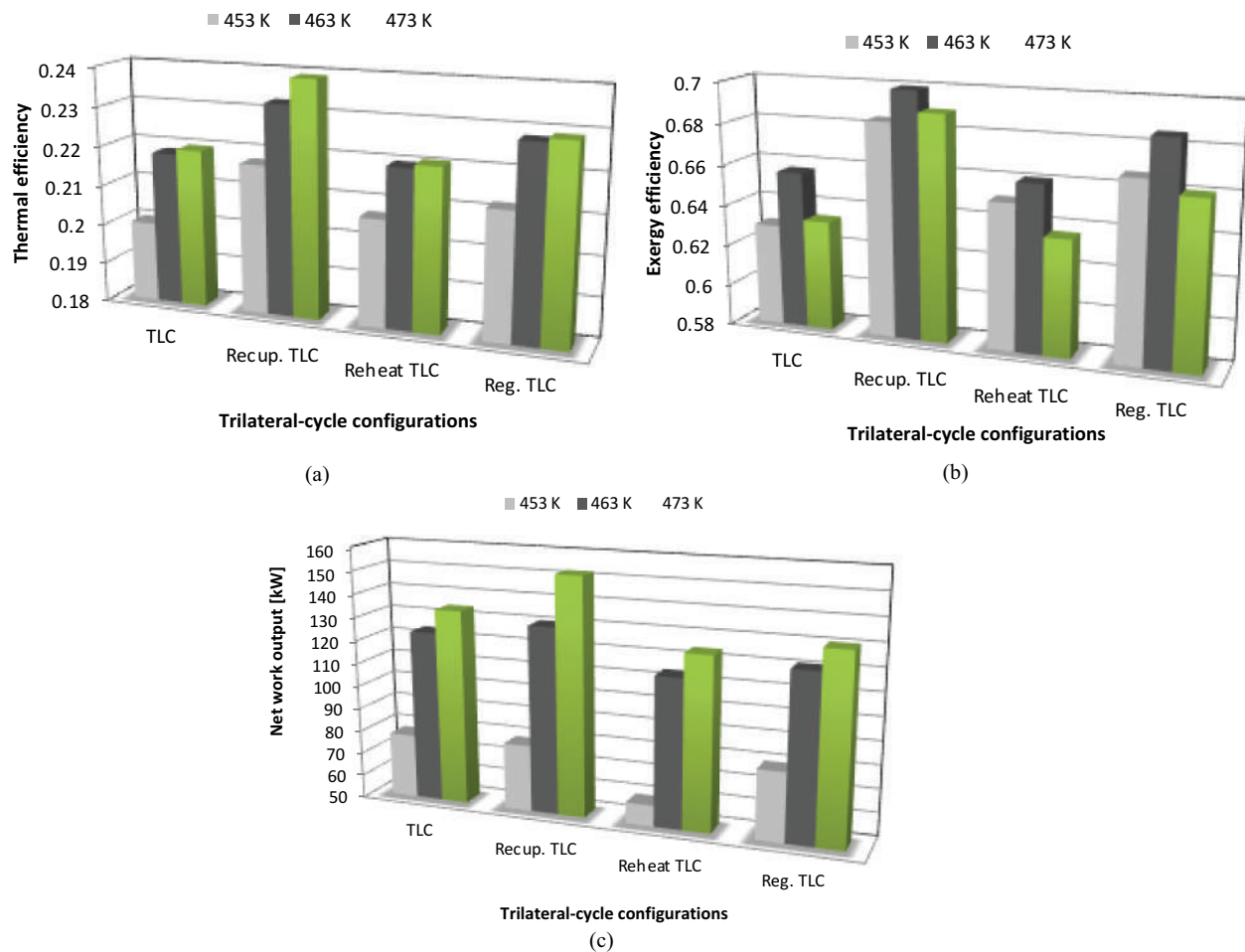


Figure 4. The TLCs' performance metrics variation with inlet temperature for (a) thermal efficiency; (b) exergy efficiency; and (c) net work output.

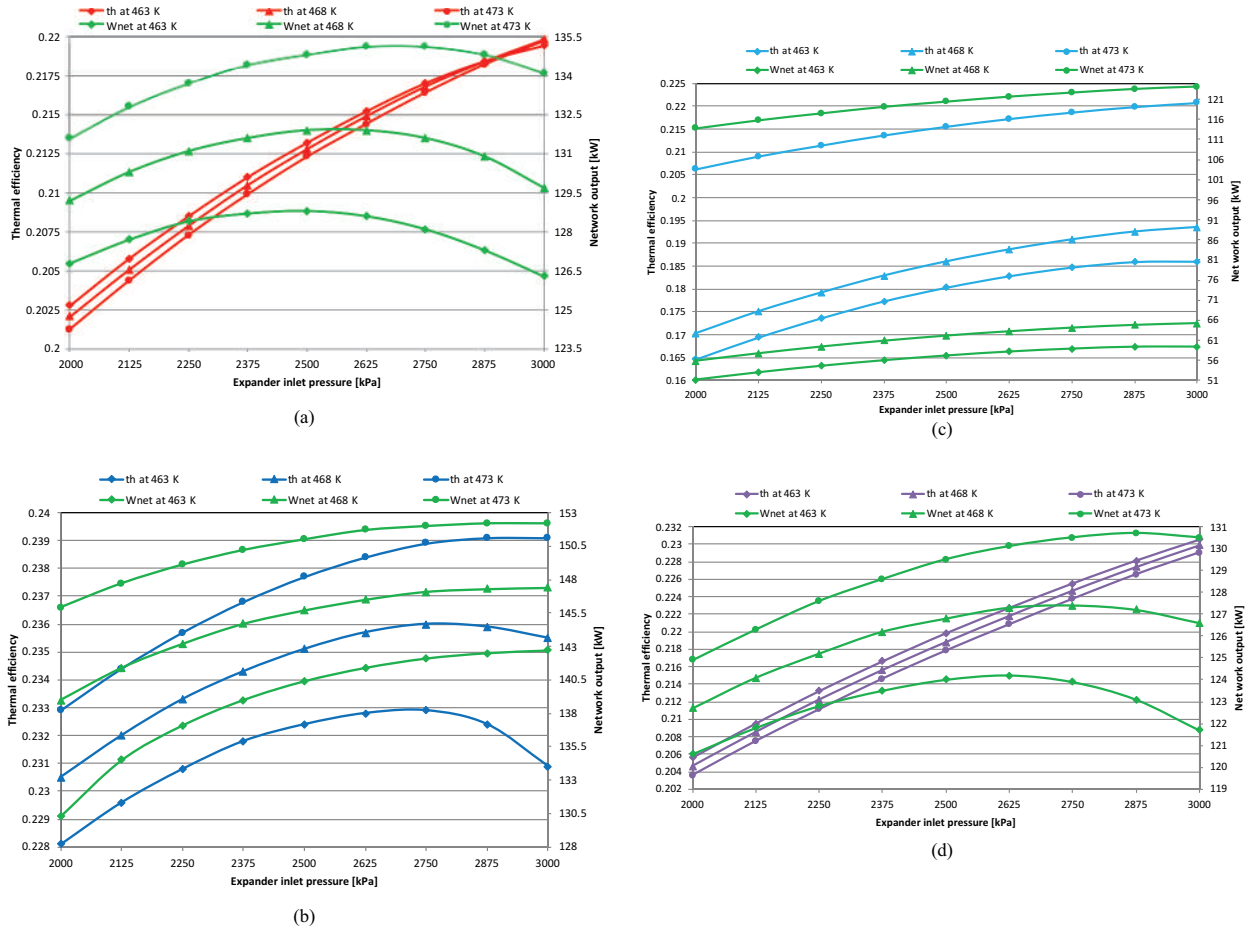


Figure 5. The thermal efficiency and net work output variation with expander inlet pressure for (a) simple TLC; (b) recuperative TLC; (c) reheat TLC; and (d) regenerative TLC.

condensers. These suggest that the thermal integration of the simple TLC configuration with recuperator and regenerator enhanced heat exchange efficiencies and the performance metrics.

Figure 4 depicts the TLCs' performance metrics variation with expander inlet temperature. Figure 4 (a) depicts that the thermal efficiencies of the simple, recuperative, reheat and regenerative TLCs varied from 20.01% to 21.97%, 21.75% to 23.91%, 20.69% to 22.07% and 21.2% to 22.9%, respectively. The increases in the thermal efficiencies were due to the significant increase in the expander work outputs as the isentropic efficiencies increased, thus, an increase in the thermal efficiencies of all the cycles. Nevertheless, the recuperative TLC still had the highest thermal efficiency with varying expander inlet temperature among the cycles. The thermal efficiency of the regenerative TLC was also higher than those of the reheat and simple TLCs. Figure 4(b) depicts that the exergy efficiencies of the simple, recuperative, reheat and regenerative TLCs varied from 62.95% to 65.66%, 68.42% to 69.96%, 63.65% to 66.08%, and 66.05% to 68.58%, respectively. Figure 4(c) depicts that the net work outputs of the simple, recuperative, reheat and regenerative TLCs varied from 77.57 to 134.1 kW, 78.57 to 152.2 kW, 59 to 124.1 kW and 79.64 to

130.5 kW, respectively. They depicted that the resulting thermal and exergy efficiencies and their corresponding net work outputs increased with an increase in inlet temperature. These increases can be attributed to the increases in total heat transfer capacity per unit mass of the working fluid injected into the expander, which increased proportionally with the expander inlet temperature.

Figure 5 depicts the TLCs' thermal efficiency and net power output variation with expander inlet pressure at the cycle high temperatures of 463 K, 468 K and 473 K for all the cycles. Figure 5(a) depicts that the thermal efficiency and net power output of the simple TLC varied from 20.13% to 21.98% and 126.8 to 135.1 kW, respectively. Figure 5(b) depicts that the thermal efficiency and net power output of the recuperative TLC varied from 22.81% to 23.91% and 130.3 to 152.2 kW, respectively. Figure 5(c) depicts that the thermal efficiency and net power output of the reheat TLC varied from 16.46% to 22.07% and 51.2 to 124.1 kW, respectively. Figure 5(d) depicts that the thermal efficiency and net power output of the regenerative TLC varied from 20.36% to 22.99% and 120.6 to 130.7 kW, respectively. They depicted that there were maximum limiting and optimum pressures at each cycle high temperature for the thermal and

exergy efficiencies and their corresponding net work outputs, respectively. The resulting thermal efficiencies of the cycles increased as their inlet pressure limit increased from 2 to 3 MPa. Their corresponding net work outputs increased with an increase in inlet pressure until respective optimum pressures were attained and then diminished with an increase in pressure. At 463 K, the thermal efficiencies of the cycles varied from 20.28% to 21.94%, 22.81% to 23.09%, 16.46% to 18.6% and 20.56% to 23.05% for the simple, recuperative, reheat and regenerative TLCs, respectively. Their corresponding net power outputs varied from 126.8 to 128.8 kW, 130.3 to 142.7 kW, 51.2 to 59.37 kW and 120.6 to 121.7 kW, respectively. While at 473 K, the thermal efficiencies of the cycles varied from 20.13% to 21.97%, 23.29% to 23.91%, 20.62% to 22.07% and 20.66% to 22.9% for the simple, recuperative, reheat and regenerative TLCs, respectively. Their corresponding net power outputs varied from 131.6 to 134.1 kW, 145.9 to 152.2 kW, 113.8 to 124.1 kW and 124.9 to 130.5 kW, respectively. These increases can be attributed to an increase in total heat transfer capacity per unit mass of the working fluid, injected into the expander, which increases proportionately with an

increase in expander inlet pressure; suggesting an optimum pressure ratio between the expander inlet and its outlet pressures.

Figure 6 depicts the TLCs' performance metrics variation with expander isentropic efficiency. Figure 6(a,b) depict that there was a proportionate increase in thermal efficiencies and an almost corresponding increase in net work output with an increase in the expander isentropic efficiency. These increases in the thermal efficiencies and net work output can be attributed to the significant increase in the specific power factor of the expander; that is, expander work output, which suggests optimum pressure ratios were attained between the expander inlets' and their outlets' pressures with an increase in isentropic efficiency. The thermal efficiencies increased from 11.21% to 22.65%, 13.07% to 25.91%, 11.66% to 23.25% and 12.76% to 25.48% for the simple, recuperative, reheat and regenerative TLCs, respectively. Their corresponding net work outputs increased from 76.41 to 147 kW, 93.58 to 152.3 kW, 57 to 135.9 kW and 73.18 to 144.4 kW respectively with an increase in expander isentropic efficiency.

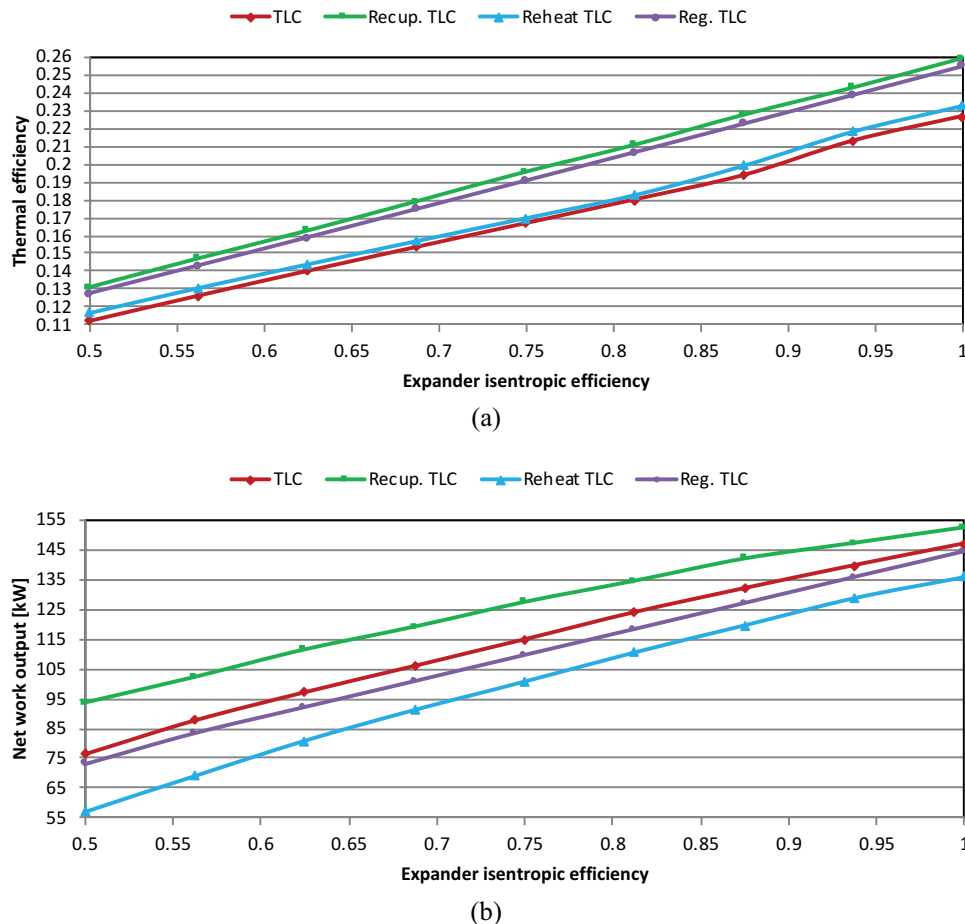


Figure 6. The TLCs' performance metrics variation with expander isentropic efficiency for (a) thermal efficiency and (b) net work output.

4. Conclusions

The thermodynamic performance analysis of the trilateral-cycle (TLC) power generator systems and the sensitivity of their system parameters on the performance metrics of the cycles were conducted using unconventional working fluid with the expander inlet temperature, expander inlet pressure and expander isentropic efficiency. The expander inlet temperatures and pressures, and its isentropic efficiencies influence the thermodynamic efficiencies and net work output of the cycles, and their effects were significant. At 473 K cycle high temperature, the thermal efficiencies of the cycles varied from 20.13% to 21.97%, 23.29% to 23.91%, 20.62% to 22.07% and 20.66% to 22.9% for the simple, recuperative, reheat and regenerative TLCs, respectively. Their corresponding net power outputs varied from 131.6 to 134.1 kW, 145.9 to 152.2 kW, 113.8 to 124.1 kW and 124.9 to 130.5 kW, respectively. The thermodynamic efficiencies and net work outputs increased with an increase in inlet temperature, inlet pressure and expander isentropic efficiency. These could be attributed to the increase in total heat transfer capacity per unit mass of the working fluid injected into the expander and its resultant increase in the specific power factor, signifying that the optimum value of pressure ratios was attained. This study would provide a valuable technique to evaluate the heat exchange performance of the TLCs, their energy conversion efficiency and thermodynamic process development.

Nomenclature

1, 2, 3, 4, 5, 6, 7 States of working fluid		Subscripts	
h	Specific enthalpy (J kg^{-1})	o	Ambient state
$\dot{m}$	Mass flow rate (kg s^{-1})	con	Condenser
P	Pressure (Pa)	cp	Condensate pump
T	Temperature (K)	cs	Cooling system
$\dot{W}$	Specific work (kW)	exp	Expander
		fp	Feed pump
Greek letters		hx	Heat exchanger
η	Efficiency (%)	opt	Optimum
		$recup$	Recuperative
Superscript		rh	Reheat
in	In/Inlet	th	Thermal
out	Out/Outlet		

Disclosure statement

No potential conflict of interest was reported by the authors.

Notes on contributors

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