

Dynamic behaviour and performance evaluation of a biomass-fired organic Rankine cycle combined heat and power (ORC-CHP) system under different control strategies

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N.B.: This is the ACCEPTED MANUSCRIPT version of this article. The final, published version of
the article can be found at: <https://doi.org/10.1016/j.applthermaleng.2024.123236>

Abstract:

The dynamic behaviour and performance evaluation of a biomass-fired organic Rankine cycle combined heat and power (ORC-CHP) system under two different control strategies are investigated in this study. The dynamic model is established, while the dynamic characteristics of key operating parameters (superheat degree, evaporating pressure and mass flow rate) with the step change of heat source temperature are examined. The effects of controlled superheat degree and controlled expander shaft work on dynamic behaviours of thermodynamic, economic and environmental performance are discussed and compared under different control methods (feedforward active control, PID passive control and combined control). Results indicate that the superheat degree is significantly influenced by the heat source temperature under the without control strategy, with the variation reaching 8.36 K. The combined control method exhibits the best control performance with overshoot and control response times of 1.23 K and 12.31 s, respectively. Compared to without control, the expander shaft

work and ORC thermal efficiency of the control strategy of superheat degree increase by 3.21 kW and 0.78%, while that of the control strategy of expander shaft work increase by 1.01 kW and 0.32%, respectively. The control strategy of superheat degree presents a significant improvement in the economic and environmental performance of the biomass-fired ORC-CHP system. This study is expected to provide better guidance for the practical operation and control of biomass-fired ORC-CHP systems.

Keywords: Biomass-fired ORC-CHP system; Dynamic behaviour; Feedforward active control; Expander shaft work; Control strategy

Nomenclature	
A	area, m ²
Bo	bond number
C_m	maintenance cost, \$
C_P	specific enthalpy, J/kg
C_V	expander coefficient
C	investment cost, \$
d	thickness, m
D	diameter, m
EA	annual return rate
Fr	froude number
g	acceleration of gravity, m/s ²
h	specific enthalpy, kJ/kg
ii	annual interest rate
L	length, m
m	mass flow rate, kg/s
n	system operating life, year
Nu	nusselt number
P	price, CNY
Pr	prandtl number
Q	heat transfer rate
r_m	system maintenance cost indicator
Re	reynolds number
top	annual running time, h
V_{cyl}	cylinder volume, m ³
Greek symbols	
ρ	density, (kg/m ³)

α	heat transfer coefficient, W/m ² ·K
β	thermal expansion coefficient
μ	density ratio
λ	thermal conductivity, W/(m·K))
u	velocity, m/s
ω	revolution speed, rpm
η_v	volumetric efficiency
η_{boil}	thermal efficiency of the biomass boiler
η_{st}	isentropic efficiency of expander
η_{sp}	isentropic efficiency of pump
η_g	motor efficiency
$\eta_{\text{ORC,th}}$	thermal efficiency
$\eta_{\text{ORC-CHP}}$	combined thermal efficiency
<i>Subscripts</i>	
bf	biomass fuel
boil	boiler
cf	cooling fluid
con	condenser
dhw	domestic hot water
e	electricity
eva1	evaporator 1
eva2	evaporator 2
exp	expander
f	fuel
g	heat source
hf	heat source fluid
p	pump

in	inlet
M	maintenance
net	net output
o	outlet
tot	total
wf	working fluid
<i>Acronyms</i>	
CHP	combined heat and power
CRF	capital recovery factor
CEPCI	chemical Engineering Plant Cost Index
CNY	chinese yuan
ECE	equivalent carbon emission
GWP	global warming potential
INV	investment
LEC	levelized electricity Cost
LHV	low-grade heat value
LQR	linear quadratic regulator
MCT	module costing technique
NAI	net annual income
NPV	net present value
ORC	organic Rankine cycle
PI	proportional-integral
PID	proportional-integral-derivative

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1. Introduction

In recent decades, the world has been facing an alarming energy crisis due to the continuous growth in global energy consumption [1]. As fossil fuels still dominate the global energy supply, there is an urgent need to explore new renewable energy to replace fossil fuels and reduce energy consumption [2]. Biomass has received increased attention due to its characteristics of high efficiency, energy savings and low carbon [3]. Moreover, the organic Rankine cycle (ORC) system has emerged as a promising technology and has been widely applied to various low-temperature heat sources [4]. The biomass-fired organic Rankine cycle combined heat and power (ORC-CHP) offers a viable solution for downsized biomass power systems [5], which can be implemented in various situations, including villages, factories, and islands.

Similar to basic ORC systems, biomass-fired ORC-CHP systems also have the drawback of low efficiency and high investment. Therefore, numerous researches devoted main efforts to improve system performance from the viewpoint of system modification, working fluid selection and parametric optimization [6]. Sikarwar et al. [7] proposed a novel biomass direct-fired ORC power generation system with a waste heat recuperator, reporting that the new system could increase the net output power by 49.93 kW. Wang et al. [8] analysed the performance of a biomass-fired ORC-CHP system using different working fluids, indicating that R134a had the best thermodynamic performance. Liu et al. [9] compared the thermodynamic performance of biomass-fired ORC-CHP systems using three different working fluids. Results showed that n-pentane had the highest ORC thermal efficiency of 16.6%. Algieri et al. [10] analysed the effects of evaporation temperature and working fluid on a biomass-fired ORC-CHP system, showing that a higher evaporation temperature ensured a better system performance, and the highest electrical efficiency of 14.6% for toluene was yielded. Zhu et al. [11] conducted an economic analysis on a biomass-fired ORC-CHP system, stating that cyclopentane owned the highest thermodynamic performance.

Stable heat source conditions are typically employed in theoretical study; however, in practical applications, heat source conditions often experience significant fluctuations due to the variability in

biomass steam generation or environmental variation [12]. This could result in the ORC system deviating from its designed operating conditions, potentially leading to the decomposition of the working fluid [13], damage to the expander due to inadequate gasification of the working fluid [14], and a decrease in expander efficiency [15]. Therefore, the dynamic behaviour of ORC systems with heat source fluctuations has received more attention. For the establishment of dynamic models of heat exchangers in ORC systems, the Finite Volume (FV) method and the Moving Boundary (MB) method were generally adopted. Although the FV method provided high accuracy, the computational speed was too slow and it was not efficient in expressing the dynamic behaviours of the ORC system [16]. The MB method could reduce the order of the heat exchanger model, which was more suitable for the analysis of dynamic behaviours and control strategies [17]. Therefore, the MB method was utilized in this study to develop the dynamic model of the heat exchanger.

To ascertain the dynamic behaviour of ORC systems under heat source fluctuations, Xu et al. [18] evaluated the ORC dynamic behaviour of the operating parameters and system performance under fluctuating conditions. Results showed that the thermal efficiency of the ORC system increased from 9.25% to 17.72% with the speed. Li et al. [19] compared the dynamic characteristics of basic ORC and latent thermal energy storage ORC (LTES-ORC) systems, showing that the LTES-ORC system could mitigate the effect of heat source fluctuations and had a larger heat source fluctuation limit compared to the basic ORC system. Yu et al. [20] analysed the effect of heat sources on the performance of basic ORC systems, reporting that the permissible relative amplitude of the heat source temperature was 25-40 K based on safety and efficiency. Wang et al. [21] established a dynamic model of heat exchangers by MB method and analysed the dynamic characteristics of four different forms of ORC systems. Shu et al. [22] examined the effect of working fluids on the dynamic response speed in natural gas engine ORC systems under the perturbation of heat source temperature and mass flow rate of working fluid, revealing that the low-temperature working fluid exhibited a faster response speed. Manuel et al. [23] compared the dynamic behaviours of a direct and indirect evaporation ORC system under fluctuating exhaust temperatures of an internal combustion engine,

indicating that direct evaporation can reduce the volume and weight of the evaporator by 70% and 88%, respectively. Although the fluctuations of heat sources are unavoidable, component damage and performance degradation caused by heat sources can be addressed by incorporating appropriate control strategies. Quoilin et al. [24] analysed the dynamic behaviour of an ORC system under fluctuations in heat source temperature and mass flow rate, showing that the system performance decreased rapidly with the decreasing heat source temperature. Therefore, three control strategies were proposed: constant evaporation temperature, optimization of evaporation temperature based on actual operating conditions and constant superheat degree. Pang et al. [25] discussed the dynamic behaviour of a 3 kW ORC system under fluctuations of heat source, showing that the control strategy of superheat degree was the best operating conditions. For the control method, Jin et al. [26] adjusted the mass flow rate of working fluid through a Proportional-Integral (PI) Controller to keep the evaporator outlet pressure constant, showing that this control method had a large impact on both the dynamic behaviour and the final steady state of the ORC system. Chen et al. [27] adopted three control methods to keep the evaporator outlet pressure constant under the fluctuation of heat source temperature, showing that the feed-forward control method presented a better control effect. Zhang et al. [28] proposed a multivariable control strategy for a waste heat recovery system by combining a linear quadratic regulator (LQR) with a PI controller. Results demonstrated that the control method had a good performance.

Looking at previous studies, the research on the dynamic behaviour and control strategies of ORC made considerable advances, but the dynamic behaviour of biomass-fired ORC-CHP systems is lacking. Furthermore, the comparison of different control methods and control strategies and the comprehensive performance evaluation of the system under control strategies have not been well reported. In this study, the dynamic behaviours and performance evaluation of the biomass-fired ORC-CHP system under two different control strategies (control strategy of superheat degree and expander shaft work) are investigated. Based on the dynamic model of a biomass-fired ORC-CHP system model, the dynamic operation characteristics of the key operating parameters (superheat

degree, evaporating pressure and mass flow rate) are discussed. Furthermore, the system thermodynamic, economic and environmental performances at different control strategies are also evaluated. To maintain the safety and stability of the output power of the system, three control methods (PID control, feedforward control and combined control) and two control strategies (control strategy of superheat degree and expander shaft work) are adopted. This study is expected to provide better guidance for the practical operation and control of biomass-fired ORC-CHP systems.

The main contributions are as follows:

(1) The dynamic model of a biomass-fired ORC-CHP system is established.

(2) The dynamic characteristics of key operating parameters (superheat degree, evaporating pressure and mass flow rate) with the step change of heat source temperature are examined.

(3) The effects of controlled superheat degree and controlled expander shaft work on dynamic behaviours of thermodynamic, economic and environmental performance are discussed and compared under different control methods (feedforward active control, PID passive control and combined control).

2. Methodology

2.1 System description

The schematic of the biomass-fired ORC-CHP system is illustrated in Fig. 1. The system mainly includes three loops: the heat source loop, the ORC loop and the domestic water loop. As shown in Fig. 1, water functions as the heat transfer fluid, facilitating the exchange of heat with the hot gas generated through biomass combustion in the heat exchanger. This process generates a water vapour at approximately 433 K. The water vapour undergoes heat exchange with evaporators 1 and 2, transferring heat to the working fluid of the ORC loop in evaporator 1 and domestic hot water in evaporator 2, respectively. Subsequently, the heat source loop is completed as the heated fluid returns to the heat exchanger.

The ORC loop comprises four primary devices: an evaporator, a condenser, an expander and a working fluid pump. The heat source transfers heat to working fluid in the evaporator1, which

transforms working fluid from a supercooled liquid to a superheated vapour, and then drives the
 expander and generator to generate electrical energy. Subsequently, the working fluid undergoes
 cooling in the condenser and then enters the pump to elevate its pressure, thereby completing the
 entire ORC loop. As for the domestic water loop, the initial step involves the domestic water entering
 the condenser to engage in heat exchange with the working fluid. Subsequently, it proceeds to the
 evaporator2, where it is further heated to 333 K by the heat source and utilized for hot water.

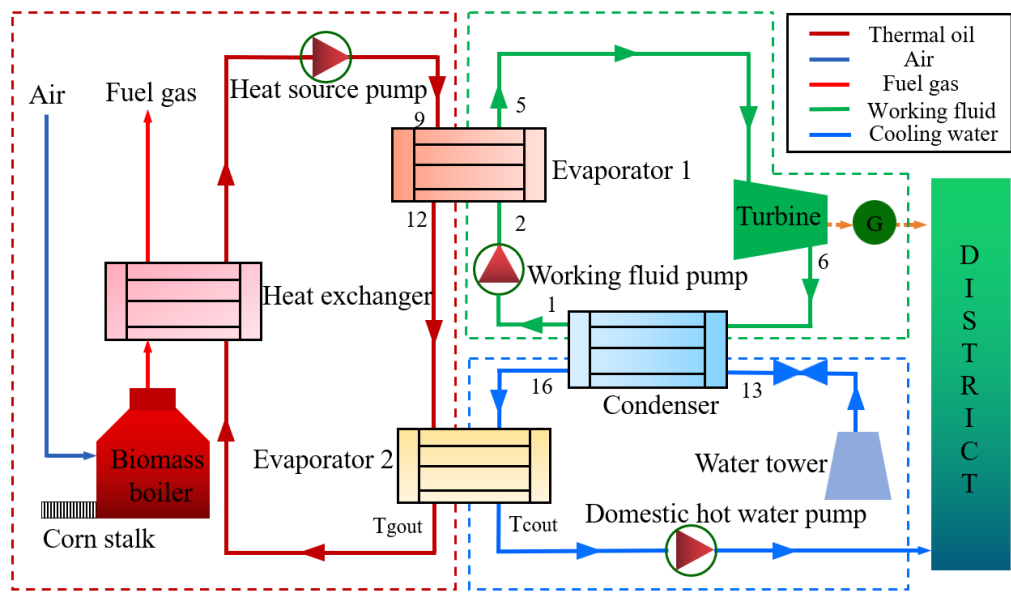


Fig. 1. Schematic of the biomass-fired ORC-CHP system

2.2 Thermodynamic model

The temperature–entropy diagram of the biomass-fired ORC-CHP system is shown in Fig. 2.
 The specific enthalpy and entropy of the working fluids are identified with NIST REFPROP.

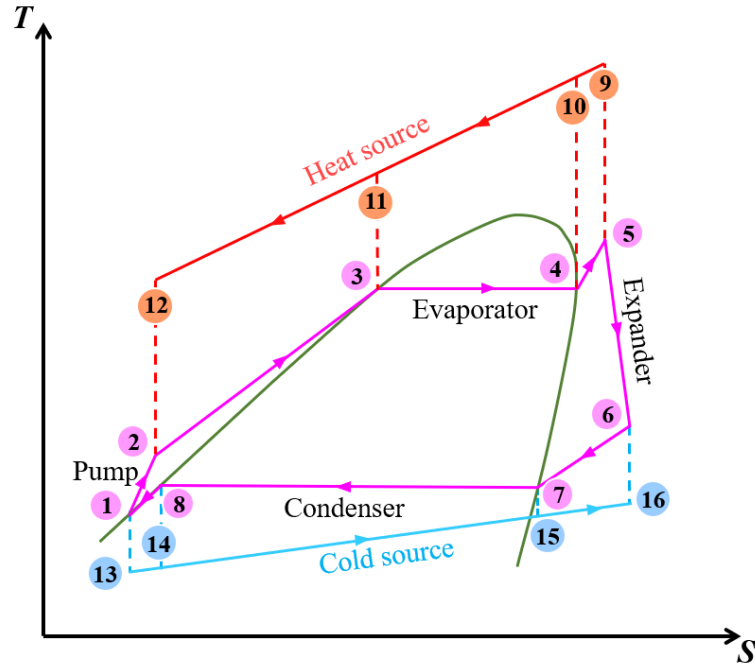


Fig. 2. Temperature–entropy diagram of the biomass-fired ORC-CHP system

Expander shaft work (W_{exp}) is constant at 30 kW, and the mass flow rate of the working fluid (m_f) is:

$$m_f = W_{\text{exp}} / (h_5 - h_6) \quad (1)$$

Heat transfer rates of the evaporator1 (Q_{eva1}), condenser (Q_{con}), and mass flow rate of cooling water (m_{cf}) are:

$$Q_{\text{eva1}} = m_f \cdot (h_5 - h_2) \quad (2)$$

$$Q_{\text{con}} = m_f \cdot (h_6 - h_1) \quad (3)$$

$$m_{\text{cf}} = Q_{\text{con}} / (h_{16} - h_{13}) \quad (4)$$

Heat transfer rate of evaporator2 (Q_{eva2}) and domestic water (Q_{dhw}) are:

$$Q_{\text{eva2}} = m_{\text{cf}} \cdot (h_{\text{dhw}} - h_{16}) \quad (5)$$

$$Q_{\text{dhw}} = m_{\text{cf}} \cdot (h_{\text{dhw}} - h_{13}) \quad (6)$$

where h_{dhw} is the specific enthalpy of water at the temperature (333K).

Shaft work of the working fluid pump (W_p), heat source pump ($W_{p,g}$) and domestic water pump ($W_{p,c}$) are:

$$W_p = m_f \cdot (h_2 - h_1) \quad (7)$$

$$W_{p,g} = 100 \cdot m_{\text{hf}} \cdot P_{\text{loss}} \cdot g \quad (8)$$

$$W_{p,c} = 100 \cdot m_{cf} \cdot P_{loss} \cdot g \quad (9)$$

where P_{loss} is the differential pressure [31], and g is the gravity acceleration of 9.8 m/s^2 .

Heat transfer rate (Q_{in}) and mass flow rate of biomass boiler (m_{bf}) are:

$$Q_{in} = (Q_{eva1} + Q_{eva2}) / \eta_{boil} \quad (10)$$

$$m_{bf} = Q_{in} / LHV \quad (11)$$

where η_{boil} is the thermal efficiency of the biomass boiler with 90% and LHV is the low-grade heat value with 17746 kJ/kg [31].

Net output work (W_{net}), thermal efficiency of ORC loop ($\eta_{ORC,th}$) and thermal efficiency of biomass-fired ORC-CHP system ($\eta_{ORC-CHP,th}$) are:

$$W_{net} = W_{exp} - W_p - W_{p,c} - W_{p,g} \quad (12)$$

$$\eta_{ORC,th} = \frac{W_{exp} - W_p}{Q_{in}} \quad (13)$$

$$\eta_{ORC-CHP,th} = \frac{W_{net} + Q_{dhw}}{Q_{in}} \quad (14)$$

2.3 Economical model

For the biomass-fired ORC-CHP system, the main investments include boilers, evaporators, condensers, expanders, and pumps. Specifically, for evaporators and condensers, a prerequisite is the computation of the heat exchange area before delving into investment cost calculations. The ORC evaporator adapts the tubular heat exchanger, while plate heat exchangers are employed for both the condenser and evaporator 2. The correlation equations for the heat transfer coefficients are shown in Table 1.

Table 1. Correlation equations for the heat transfer coefficients

Heat exchanger	Side	heat transfer coefficients
Tubular heat exchanger	Outside tube	$\alpha_{h,i} = 0.023 \cdot \frac{\lambda}{d} \cdot Re^{0.8} \cdot Pr^{0.4}$
	Inside tube (single phase)	$\alpha_i = 0.027 \cdot \frac{\lambda}{d} \cdot Re^{0.8} \cdot Pr^{0.4} \cdot \left(\frac{\eta_f}{\eta_w}\right)^{0.14}$
	Inside tube (two phase)	$\frac{\alpha_i}{\alpha_0} = 1.136 \cdot D_0^{-0.9} \cdot (25Fr_0)^{0.3} + 667.2 \cdot Bo^{0.7} \cdot F_{fl}$
	Outside tube	$Nu = 4.118 \cdot Re^{0.4} \cdot Pr^{1/3}$

Plate heat
exchanger

Inside tube

$$Nu = 0.724 \cdot \left(\frac{6\beta}{\pi}\right)^{0.646} \cdot Re^{0.583} \cdot Pr^{1/3}$$

Total heat transfer coefficients (α_i) of plate heat exchanger and tubular heat exchanger are:

$$\frac{1}{\alpha_i} = \frac{1}{\alpha_{c,i}} + \frac{1}{\alpha_{h,i}} + \frac{d}{\lambda} \quad (15)$$

$$\frac{1}{\alpha_i} = \frac{1}{\alpha_{i,i}} \cdot \frac{A_o}{A_i} + \frac{D_o}{2\lambda} \cdot \ln \frac{D_o}{D_i} \cdot \frac{1}{\alpha_{c,i}} + \frac{1}{\alpha_{o,i} \eta_f} \cdot \frac{A_o}{A_i} \quad (16)$$

where $\alpha_{c,i}$ and $\alpha_{h,i}$ are the heat transfer coefficients of the heat source fluid and the cooling water, respectively; λ and d are the thermal conductivity and the heat exchanger tube thickness, respectively; $\alpha_{i,i}$ and $\alpha_{o,i}$ are the internal and outer of heat exchanger tube heat transfer coefficients for the working fluid, respectively; A_i and A_o are the internal and outer surface areas of the heat exchanger tube, respectively; η_f is the efficiency of fins.

Heat exchange area for different section (A_i) is:

$$A_i = Q_i / \alpha_i / \Delta T_i \quad (17)$$

where Q_i and ΔT_i are the heat transfer rate and temperature difference in different phase regions, respectively.

In this study, the module costing technique (*MCT*) is used to estimate the equipment cost, and the equipment investment is:

$$\lg C_{p,X} = K_{1,X} + K_{2,X} \cdot \lg Y + K_{3,X} \cdot (\lg Y)^2 \quad (18)$$

where $C_{p,X}$ and Y are the basic investment of component and the relevant parameters of each component of the biomass-fired ORC-CHP system, respectively. The specific module costs and parameters of the components are shown in Table 2.

Table 2. Cost calculation coefficient of each component.

Component	Models	$K_1/K_2/K_3$	$C_1/C_2/C_3$	F_M	F_{BM}
Heat exchangers	$C_{BM,X} = C_{p,X} \cdot F_{BM,X}$	3.214	-0.065		
	$\lg C_{p,X} = K_{1,X} + K_{2,X} \cdot \lg A + K_{3,X} \cdot (\lg A)^2$	0.269	0.050	1.25	-
	$\lg F_{p,X} = C_{1,X} + C_{2,X} \cdot \lg P + C_{3,X} \cdot (\lg P)^2$	0.054	0.015		
Expanders	$C_{BM,X} = C_{p,X} \cdot (F_{BM,X} \cdot F_{p,X})$	3.514	-	-	3.5

	$\lg C_{p,X} = K_{1,X} + K_{2,X} \cdot \lg A + K_{3,X} \cdot (\lg W)^2$	0.589	-		
	$\lg F_{p,X} = C_{1,X} + C_{2,X} \cdot \lg P + C_{3,X} \cdot (\lg P)^2$	0	-		
	$C_{BM,X} = C_{p,X} \cdot F_{BM,X}$	3.389	-0.394		
Pump	$\lg C_{p,X} = K_{1,X} + K_{2,X} \cdot \lg A + K_{3,X} \cdot (\lg W)^2$	0.054	0.396	1.5	-
	$\lg F_{p,X} = C_{1,X} + C_{2,X} \cdot \lg P + C_{3,X} \cdot (\lg P)^2$	0.15	-0.002		

Costs of revised equipment investment (C_X) and working fluid (C_{wf}), annual operating costs of biomass fuel (C_{bf}) and biomass boilers (C_{boiler}), and system maintenance costs (C_m) are:

$$C_X = C_{BN,X} \cdot CEPCI_{2020} / CEPCI_{2001} \quad (19)$$

$$C_{wf} = (C_{exp} + C_{pump} + C_{con} + C_{eva1}) \cdot r_{wf} \quad (20)$$

$$C_{bf} = m_{bf} \cdot top \cdot P_{bio} \quad (21)$$

$$C_{boiler} = P_{boil} \cdot Q_{in} / 10^6 \cdot CEPCI_{2020} / CEPCI_{2018} \quad (22)$$

where $C_{BN,X}$ is investment of components, r_{wf} is the percentage of working fluid cost and takes the value of 2% [30], and $CEPCI$ is the chemical engineering plant cost index.

Total system investment (INV_{tot}) and maintenance costs (C_m) are:

$$INV_{tot} = C_{exp} + C_{eva1} + C_{eva2} + C_{con} + C_{pump} + C_{pc} + C_{pg} + C_{boiler} + C_{wf} + C_{bf} \quad (23)$$

$$C_m = r_m \cdot INV_{tot} \quad (24)$$

where r_m is system maintenance cost indicator.

Annual return on electricity generation (EA_e) and domestic hot water (EA_w) are:

$$EA_e = W_{net} \cdot top \cdot P_e \quad (25)$$

$$EA_w = m_{dhw} \cdot top \cdot P_w \quad (26)$$

where P_e and P_w are the price of electricity and domestic hot water, respectively.

Capital recovery cost (CRF) and levelized electricity cost (LEC) of system are:

$$CRF = ii \frac{(1+ii)^n}{(1+ii)^n - 1} \quad (27)$$

$$LEC = \frac{(CRF \times \dot{C}_{tot} + \dot{C}_{bf} + \dot{C}_m)}{W_{net} \cdot top} \quad (28)$$

where ii is the annual interest rate of 5% [29].

Net annual income (NAI) and the net present value (NPV) are:

$$NAI = EA_e + EA_w - C_m - C_{bf} \quad (29)$$

$$NPV = \sum_{n=1}^{n=20} \frac{NAI}{(1+i)^n} - C_{tot} \quad (30)$$

where n is the system operating life of 20 years. The specific index parameters are shown in Table 3.

Table 3. Economic parameters of the biomass-fired ORC-CHP system.

Parameter	Value
Chemical engineering plant cost index in 2001 $CEPCI_{2001}$	397
Chemical engineering plant cost index in 2018 $CEPCI_{2018}$	603.1
Chemical engineering plant cost index in 2020 $CEPCI_{2020}$	596.2
Cost of biomass boiler P_{boil} (CNY/MWth)	12×10^4
Price of biomass fuel P_{bio} (CNY/t)	500
Price of electricity P_e (CNY/kWh)	0.87
Price of domestic hot water P_w (CNY/t)	25
Cost of system maintenance r_M	1.65
Annual running time top (h)	8000
System operating life n (years)	20

2.4 Environmental model

In this study, Equivalent Carbon Emission (ECE) is adopted as the environmental evaluation index of the biomass-fired ORC-CHP system. The indicator of ECE mainly consists of direct emissions and indirect emissions. The direct emissions include leakage and losses of working fluids and heat source emissions, while the indirect emissions are mainly caused by energy consumption, construction phase and recovery phase losses [32]. In this study, the butane is chosen as the working fluid because of its excellent physical and environmental properties, which the ODP and GWP are 0 and 20, respectively. Furthermore, the butane has a lower cost compared to other organic working fluids, hence lowering the total cost of the system. The specific properties are shown in Table 4.

Table 4. Properties of working fluids.

Working Fluid	Critical Pressure (MPa)	Critical Temperature (K)	molecular mass	ODP	GWP
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Butane (R600)	3.796	425.01	58.12	0	20
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Equivalent Carbon Emission (ECE) is:

$$ECE = FC \cdot (nALR + EOL) \cdot GWP + m_{CM} + m_r CMR \quad (31)$$

where FC is a work mass charge of 5.57 kg/kW, GWP and ALR are working fluid global warming potential values and annual leakage rates of 20 and 2%, respectively. n and EOL are the system operating life and the loss rate of working fluid at the end, which are 20 and 10%, respectively; m and m_r are the mass of non-renewable material and renewable material [33]. CM and CMR are the CO_2 production of per unit mass non-renewable and renewable material in manufacture and disposal processes [34]. The material mass for various components is presented in Table 5, and Table 6 provides the information of material manufacturing and disposal emissions.

Table 5. Equations of material mass for relevant components

Component	Material mass
Evaporate	$m_{eva} = \rho V = \rho \cdot \left[\frac{\pi(d_o^2 - d_i^2)}{4} \cdot \frac{A_{eva}}{\pi d_o} + \frac{L}{Y} \cdot \frac{\pi(d_t^2 - d_i^2)\delta_f}{4} \right]$
Condenser	$m_{cond} = \rho \cdot \frac{\pi(d_o^2 - d_i^2)}{4} \cdot \frac{A_{cond}}{\pi d_o}$
Expander	$m_{tur} = 31.22 \cdot W_{net}$
Pump	$m_{pump} = 14.0 \cdot W_{pump}$

Table 6. Manufacturing and disposal emissions of material

Material	Virgin material manufacturing emissions (kgCO ₂ , eq/kg)	Mixed material manufacturing emissions (kgCO ₂ , eq/kg)	Disposal emissions (kgCO ₂ , eq/kg)
Steel	1.80	1.43	0.07
Aluminum	12.6	4.50	0.07
Copper	3.0	1.64	0.07
Plastics	2.8	2.61	0.07

3 Dynamic model and control strategies

3.1 Dynamic model of heat source temperature

The heat from the combustion of biomass is transferred to the heat transfer fluid (water) via the flue gas in biomass boiler, and then the water vapour is directed to an ORC evaporator to vaporize the organic working fluid. However, because of unstable mass flow rates of biomass and heat transfer fluids, the heat source also exhibits fluctuations in temperature. In this study, the inlet temperature of the fluctuating heat source is assumed as a step function in the following equation:

$$T_h = 160 \cdot (1 \pm SCR) + 273 \quad (32)$$

where SCR is the step change rate (6%, 4% and 2% is selected in this study).

3.2 Dynamic model of the biomass-fired ORC-CHP system

3.2.1 Dynamic models for evaporator and condenser

The methods for constructing dynamic models of heat exchangers mainly include the discrete method and the moving boundary (MB) method. In this paper, the MB method is chosen due to its high accuracy, reliability, and low computational workload. The schematic of three regions is depicted in Fig. 3. The essence of the MB model lies in dynamically monitoring the lengths of the three-phase regions (subcooling, two-phase, and superheating regions), determining the heat transfer coefficients, and subsequently solving for the desired values using energy and mass equations. The more details can refer to the literature [35].

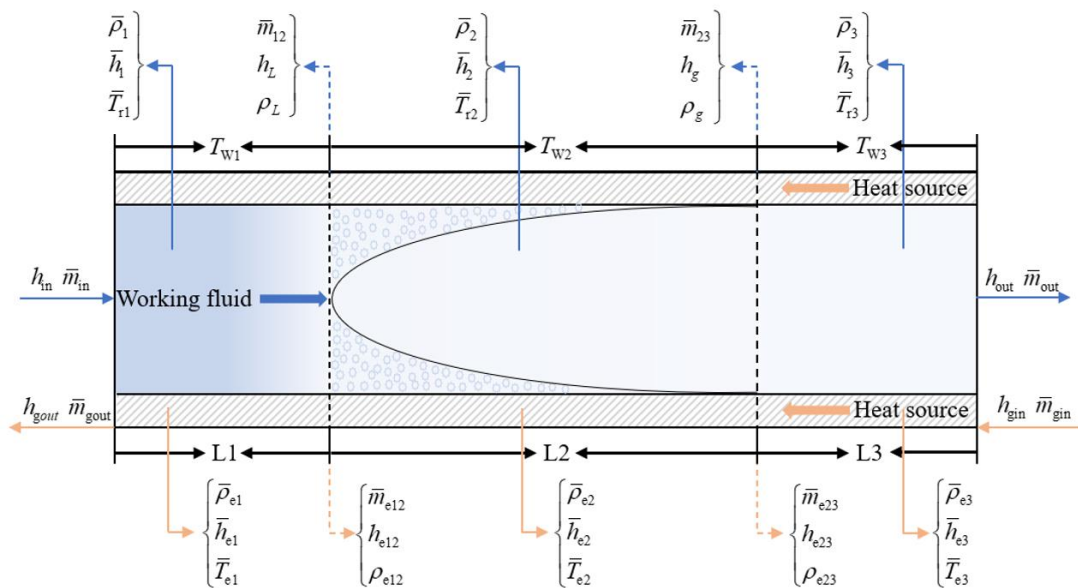


Fig. 3. Schematic of the MB model

The general differential mass balance and energy balance equations for the three regions, along with the simplified energy balance equation for the wall are:

$$\int_0^{L1} \frac{\partial(A\rho)}{\partial t} dz + \int_0^{L1} \frac{\partial \bar{m}}{\partial z} dz = 0 \quad (33)$$

$$\int_0^{L1} \frac{\partial(A\rho h - A p)}{\partial t} dz + \int_0^{L1} \frac{\partial \bar{m} h}{\partial z} dz = \int_0^{L1} \alpha_i \pi D_i (T_w - T_r) dz \quad (34)$$

$$C_{pw} \rho_w A_w \frac{dT_w}{dt} = \alpha_i \pi D_i (T_r - T_w) + \alpha_o \pi D_o (T_a - T_w) \quad (35)$$

The phase-change heat exchanger MB model is obtained by integrating the aforementioned equations (33)-(35) for each of the three regions ($L1, L2, L3$) respectively. The dynamic modelling of the condenser follows a similar approach to that of the evaporator, utilizing the MB method. The specific design parameters of the heat exchanger are detailed in Table 7, with no phase transition occurring in evaporator 2.

Table 7. Specific design parameters of the heat exchangers.

Heat exchanger		Design parameters				
Evaporator 1	$L1/L2/L3$ (m)	$T_{g,in}$ (K)	$T_{g,out}$ (K)	$T_{f,in}$ (K)	$T_{f,out}$ (K)	Q_{eva1} (kW)
	33.9/20.4/13.3	433	426.82	299.04	408	213.68
Condenser	$L1/L2/L3$ (m)	$T_{cf,in}$ (K)	$T_{cf,out}$ (K)	$T_{f,in}$ (K)	$T_{f,out}$ (K)	Q_{con} (kW)
	1.92/8.36/2.05	298	304.02	347.92	308	185.26
Evaporator 2	L (m)	$T_{g,in}$ (K)	$T_{g,out}$ (K)	$T_{cf,in}$ (K)	$T_{cf,out}$ (K)	Q_{eva2} (kW)
	28.18	426.825	400.78	304.02	333	891.58

3.2.2 Dynamic models for the pump and expander

Displacement pump is adopted in the study, and the mass flow rate of pump (m_p) is:

$$m_p = \eta_v \cdot \rho_{p,in} \cdot V_{cyl} \cdot \omega \quad (36)$$

where η_v , V_{cyl} , and ω are the volumetric efficiency, cylinder volume, and pump speed, respectively [33].

Shaft work of pump (W_p) and the enthalpy at pump outlet ($h_{p,out}$) are:

$$W_p = m_p \cdot (h_{p,out} - h_{p,in}) / \eta_g \quad (37)$$

$$h_{p,out} = h_{p,in} + (h_{p,isout} - h_{p,in}) \cdot \eta_{sp} \quad (38)$$

where η_g and η_{sp} are the isentropic pump efficiency and motor efficiency respectively, set to 0.7 and 0.85 respectively. $h_{p,isout}$ is the ideal outlet specific enthalpy of the pump for isentropic process.

Mass flow rate (m_{exp}), shaft work (W_{exp}) of expander and the specific enthalpy of expander outlet ($h_{exp,out}$) are:

$$m_{exp} = C_v \sqrt{\rho_{exp,in} P} \quad (39)$$

$$W_{exp} = m_{exp} \cdot (h_{exp,in} - h_{exp,out}) \cdot \eta_g \quad (40)$$

$$h_{exp,out} = h_{exp,in} - (h_{exp,in} - h_{exp,isout}) \cdot \eta_{st} \quad (41)$$

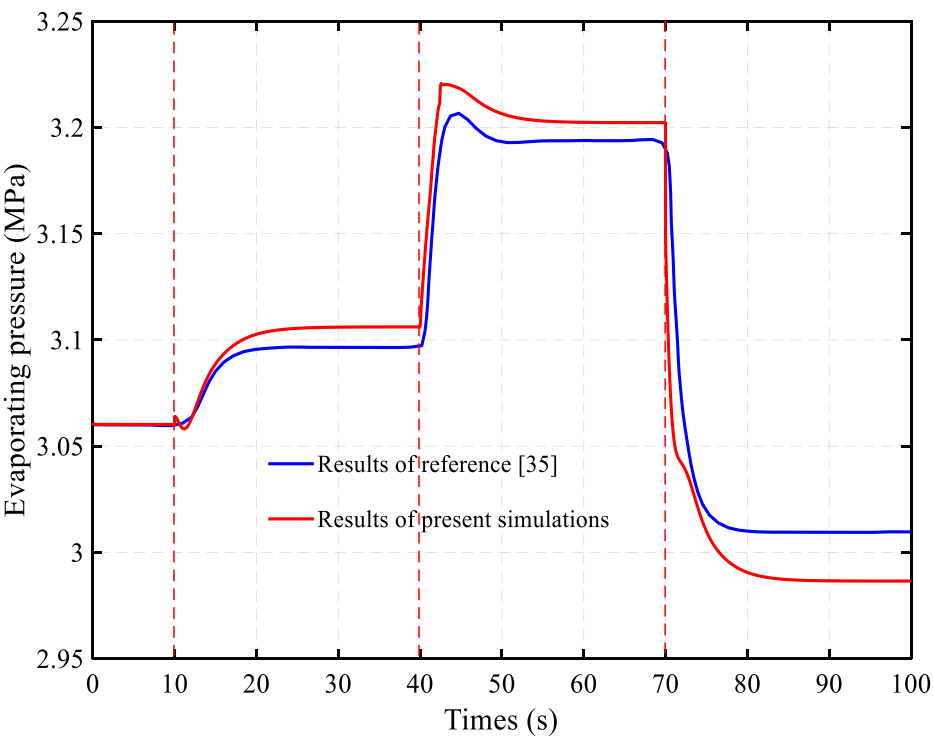
where C_v and P are the nozzle coefficient and the evaporating pressure, respectively. $h_{exp,isout}$ and η_{st} are the ideal outlet enthalpy of the expander for isentropic process and expander efficiency of 0.85, respectively [36].

During the modelling of each system component, it's crucial to verify the accuracy of each part against the design conditions, with particular emphasis on the heat exchanger to ensure that the output of the dynamic model aligns with the design specifications. Once all components are established, the dynamic model of the biomass-fired ORC-CHP system can be derived by sequentially connecting each component.

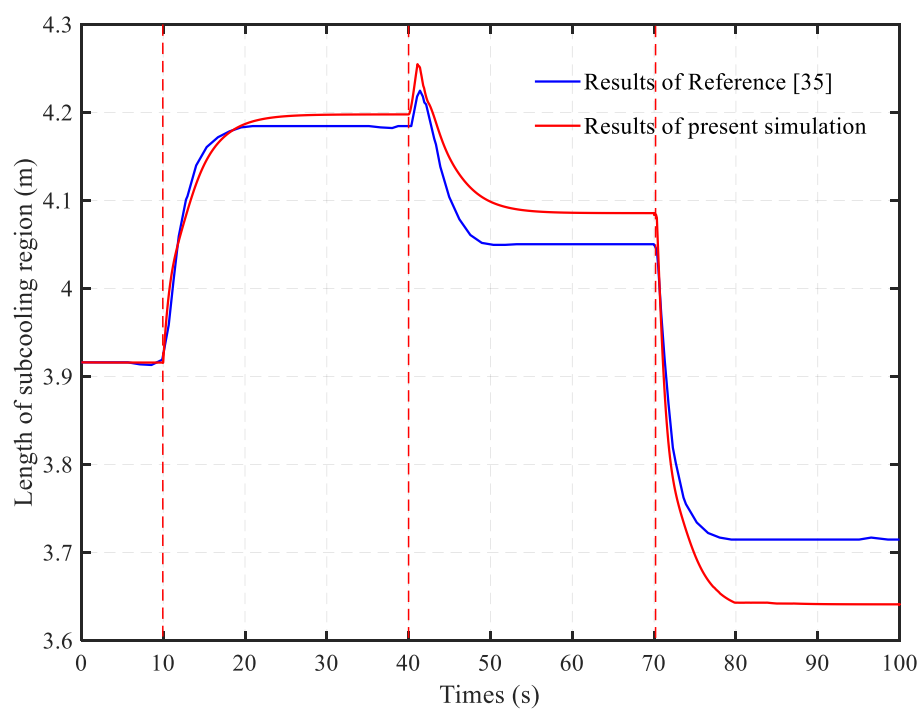
3.3 Model validation

The comparison of simulation results between this study and Ref. [35] is depicted in Fig. 4. At 10 s, the pump speed is increased by 5%, followed by a 10% increase in the external heat transfer coefficient at 40 s, and a 10% increase in the nozzle coefficient at 70 s. As shown in Fig. 4(a) and Fig. 4(b), the evaporating pressure and the length of subcooling region keep increasing with pump speed ($t=10$ s), which is expected. The increase of external heat transfer coefficient ($t=40$ s) intensifies the heat transfer effect of the working fluid in the evaporator, resulting in the decrease of the length of the subcooling region and the increase of the evaporating pressure. A larger nozzle coefficient ($t=70$ s) leads to a decrease in the temperature of the working fluid in the evaporator, which leads to a decrease in the length of subcooling region of the evaporating pressure. Furthermore, it can be seen

289 that the dynamic model of this study is in sufficient agreement with the model in Ref. [35], and it
290 could be used for subsequent research.



(a)



(b)

Fig. 4. Model validation. (a) Evaporating pressure. (b) Length of subcooling region.

3.4 Control Strategies

The fluctuation of the heat source has a significant effect on the operating conditions and system performance. Therefore, it is necessary to stabilize the system operating conditions by applying control methods in non-stationary heat source conditions. The control method mainly includes the active control according to the input heat source conditions, and the passive control according to the feedback signal of the system operation condition. Three control methods are selected in this study, which are static feedforward active control, PID passive control, and the combination of both.

In this study, to ensure the security of the biomass-fired ORC-CHP system involves preventing the ingress of the liquid working fluid into the expander and the decomposition of the working fluid, the superheat degree of the evaporator outlet (T_{sup}) is adopted as the control target parameter. Furthermore, maintaining the stability of the system's output power requires controlling the expander shaft work (W_{exp}), with the pump speed (ω) serving as the controlled input variable.

The logic of the three control methods is shown in Fig. 5. The feedforward active control strategy involves computing the control input for various heat source temperatures using the heat transfer equilibrium principle, thereby obtaining the MAP between heat source temperatures and the control input. As the heat source temperatures varies, real-time adjustments to the control input are made based on this MAP to achieve the desired control effect [24]. The second control method is based on proportional-integral-derivative (PID) control, and it is implemented by utilizing the PID Controller module in Simulink. The PID controller reads the difference between the control variable and the control target in real time, and adjusts the control input variable through the superposition of proportional, integral and differential links, then adjusts the control variable to the control target [36]. In this study, the PID control method adopts the real-time T_{sup} and W_{exp} as the feedback variable, and controls the ω by the PID controller. The third control method is a combination of PID control and static feedforward control, the control input is still based on the MAP of the static feedforward control, but adds the second correction of the PID controller to the control input.

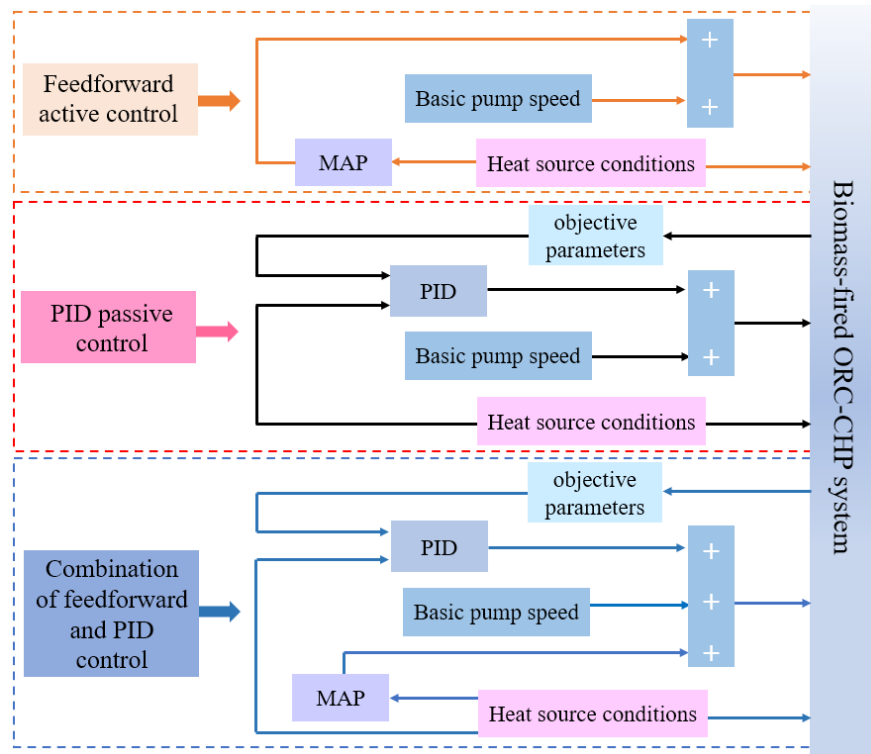


Fig. 5. Logic of three control methods

4 Results and discussion

4.1 Dynamic characteristics of step change working conditions

The dynamic behaviour of the biomass-fired ORC-CHP system is significantly affected by the heat sources characteristic. Therefore, it is significant to investigate the effect of heat source characteristic on the dynamic behaviour to evaluate the system adaptability. In this section, the effects of T_h on system operating parameters (T_{sup} , P_{eva} , m_f), the thermodynamic performance indicators (W_{exp} , W_{net} , $\eta_{ORC,th}$, $\eta_{ORC-CHP,th}$), environmental and economic characteristics (ECE , LEC , NPV) are investigated. The heat source characteristics (T_h) maintain the design working condition between 0-50 s, rise according to the step change ratio (SCR) from 50-150 s, return to the design working condition from 150-250 s, and finally decrease from 250-350 s. The SCR of T_h are 2%, 4% and 6%, respectively. In order to evaluate the system dynamic response speed, the rise time and settling time are adopted, while the shorter time represents the faster response speed. The rise time is the time required for the response to rise from 10%-90 % of its final value in overdamped systems (since most of the parameters in this study vary drastically at the initial changes of heat source, it approximates from 0%-90%), and 0%-100% for underdamped systems. The settling time is the time required for

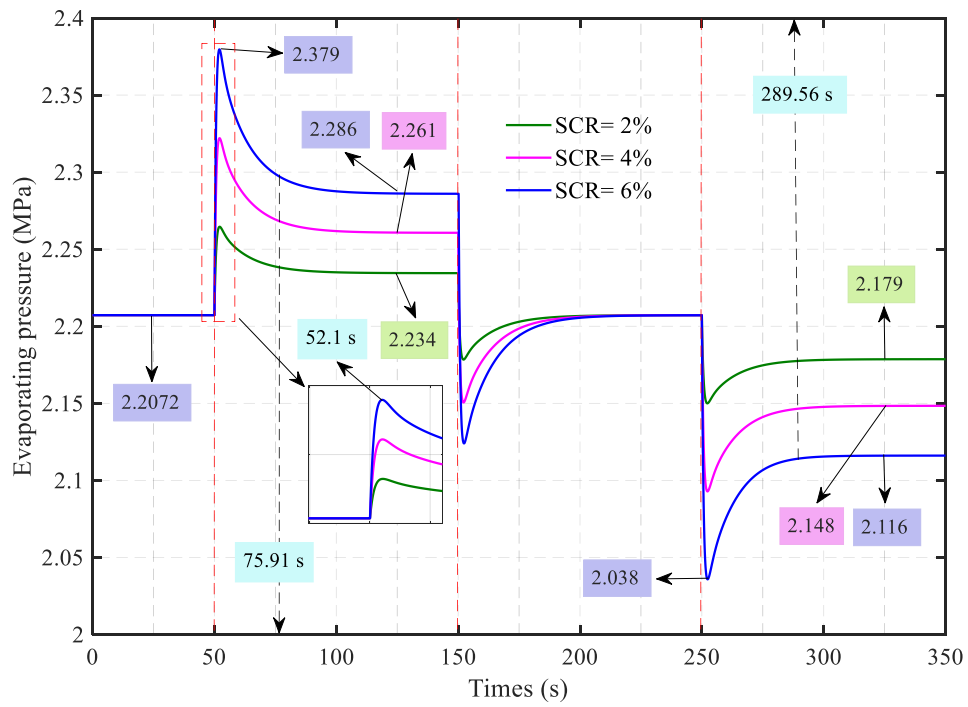
the response to reach and stay within a range of certain percentage (2%) of the final value [37]. The main design working conditions of the biomass-fired ORC-CHP system is shown in Table 8.

Table 8. Main design parameters of the biomass-fired ORC-CHP system.

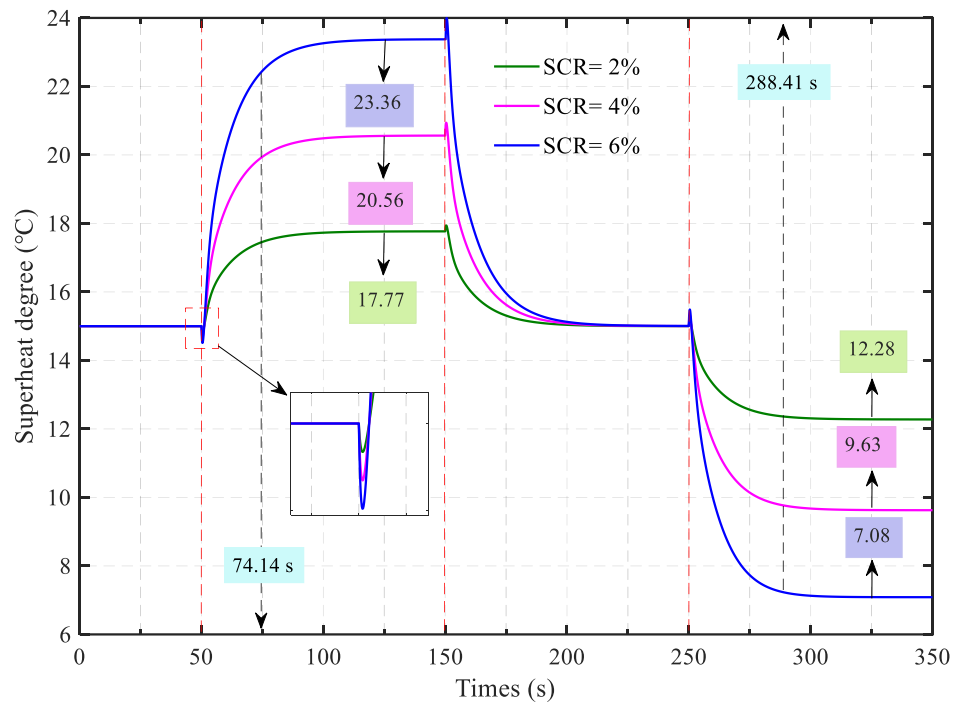
The parameters of the biomass-fired ORC-CHP system	Value
Expander shaft work W_{net}	30 kW
Domestic hot water outlet temperature T_{dhw}	333 K
Heat source inlet temperature T_9	433 K
Evaporating temperature T_{eva}	393 K
Evaporating pressure P_{eva}	2.21 MPa
Superheat degree T_{sup}	15 K
Condensing temperature T_{con}	308 K
Isentropic efficiency of expander η_{st}	0.85
Isentropic efficiency of pump η_{sp}	0.85
Pressure loss of domestic hot water pumps and boiler pumps P_{loss}	0.15 MPa
Mass flow rate of heat source m_{hf}	8 kg/s
Mass flow rate of working fluid m_{wf}	0.41 kg/s
Mass flow rate of domestic water m_{cf}	7.36 kg/s

Figure 6 shows the effect of heat source temperature (T_h) on the dynamic characteristics of superheat degree (T_{sup}), evaporating pressure (P_{eva}) and mass flow rate of working fluid (m_f). As shown in Fig. 6(a), in the step-up phase of 50-150 s, the P_{eva} exhibits a tendency of first drastic increasing and then decreasing with T_h , while the increase time is about 2.1 s. It is prevalent due to the response delay inside the evaporator, and it is negatively related to the heat transfer coefficient of the heat source. Furthermore, with the step increase of T_h , the heat transfer rate of the evaporator (Q_{eva1}) keeps increasing with the average temperature of the heat source, resulting in the increase in P_{eva} . However, with the decrease in heat source temperature at the evaporator outlet, the Q_{eva1} gradually increases, causing the P_{eva} to start to slow down. As shown in Fig. 6(b) and Fig. 6(c), the T_{sup} and m_f present the same trend during 50-150s, with an increase first and then stabilization. It is worth noting that the T_{sup} exhibits a downward fluctuation at the beginning due to the sharp rise of P_{eva} , but keeps increasing with the decrease of P_{eva} . From Eq. (36), the m_f is mainly influenced by the density of the working fluid at the pump inlet. Moreover, the heat transfer rate of the condenser keeps

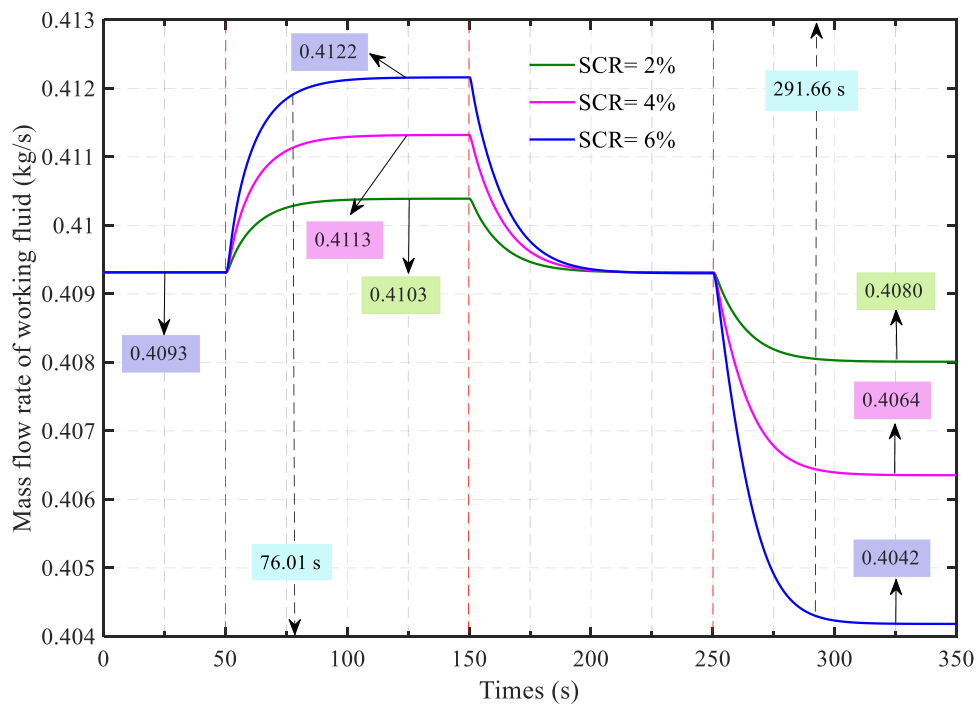
increasing with Q_{eva1} , leading to a decrease in the enthalpy of the working fluid at the pump inlet, while the m_f presents an increasing trend with T_h . Similar to the rising phase of the T_h , the P_{eva} , T_{sup} and m_f exhibit the opposite trend when the T_h decreases. Meanwhile, when the SCR is 6%, 4% and 2%, the relative average variations of the P_{eva} are 3.55%, 2.44% and 1.21%, while that of T_{sup} are 57.3%, 37.1% and 18.5%, respectively. The relative average variations of m_f are the least, with 0.708%, 0.489% and 0.244%, respectively. From the above findings, it can be found that the T_h has the greatest effect on the T_{sup} , and the average variation of T_{sup} is 8.14 °C. It is a broad swing and required to pay extra attention to the damage on the expander and the working fluid when the T_{sup} is too low or too high in actual operation. Therefore, the control strategy of T_{sup} is essential to ensure the security of the biomass-fired ORC-CHP system. In addition, it also can be found that the rise time of P_{eva} , T_{sup} and m_f are 25.91 s, 21.14 s and 26.01 s, and the settling time are 39.56 s, 38.41 s and 41.66 s, respectively. The T_{sup} has the fastest response speed, whereas the m_f yields the slowest response speed.



(a)



(b)

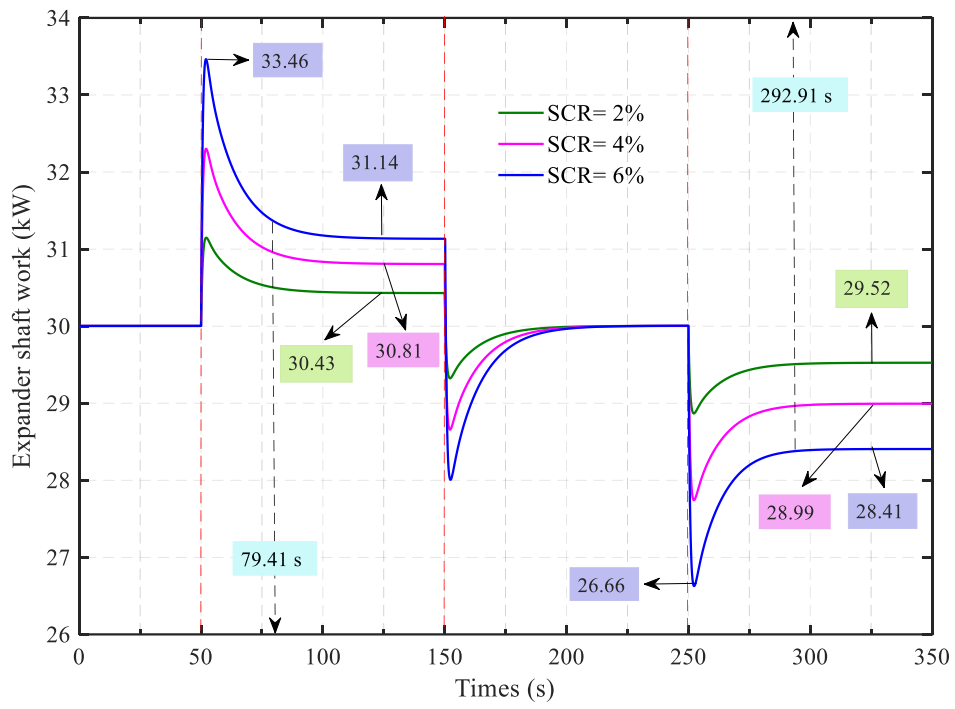


(c)

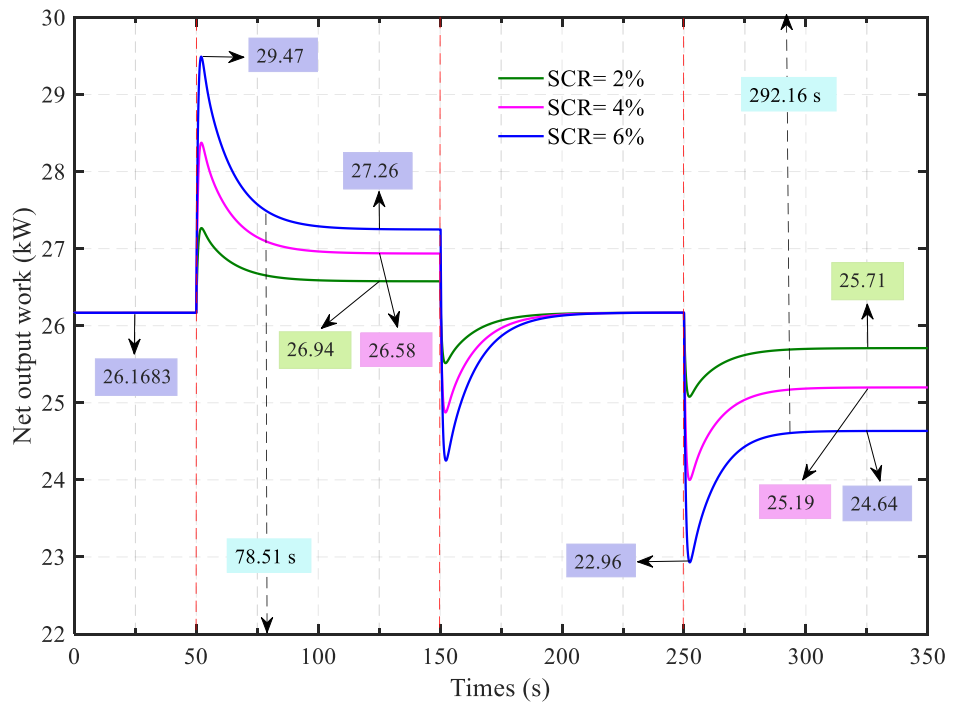
Fig. 6. Variations of key operating parameters with heat source temperature. (a) Evaporating pressure. (b) Superheat degree of evaporator outlet. (c) Mass flow rate.

The effect of T_h on the dynamic characteristics of W_{exp} , W_{net} , $\eta_{\text{ORC,th}}$ and $\eta_{\text{ORC-CHP,th}}$ is shown in Fig. 7. For these thermodynamic indicators, the W_{exp} and W_{net} determine the generating capacity and production capacity of the biomass-fired ORC-CHP system, while the $\eta_{\text{ORC,th}}$ and $\eta_{\text{ORC-CHP,th}}$ ascertain the energy utilization efficiency of the ORC loop and the whole system, respectively. As shown in Fig. 7(a), it can be seen that the trend of the W_{exp} with T_h is similar to the P_{eva} , with both exhibiting negative damping characteristics. The W_{exp} are mainly affected by the mass flow rate of working fluid in the expander and enthalpy difference. With the rise of the T_h at 50 s, the P_{eva} maintains a rapid increase, resulting in the increase of the mass flow rate of the expander, which leads to the increase of the W_{exp} . Moreover, with the decrease in P_{eva} and the increase in inlet enthalpy of the expander, the expander shaft work begins to decrease. From Fig. 6(c), it can be found that the T_h has the least effect on the m_f , so the shaft work of pump varies relatively slightly, leading to the similar trend of the W_{exp} and the W_{net} . From Fig. 7(c), the $\eta_{\text{ORC,th}}$ presents a decreasing trend with T_h , which may be attributed to the fact that the increment of Q_{eva1} is much larger than the W_{net} . Different from the $\eta_{\text{ORC,th}}$, the heat transfer rate of domestic water (Q_{dhw}) also affects the $\eta_{\text{ORC-CHP,th}}$, while Q_{dhw} keeps increasing with the T_h , resulting in the increase in the $\eta_{\text{ORC-CHP,th}}$.

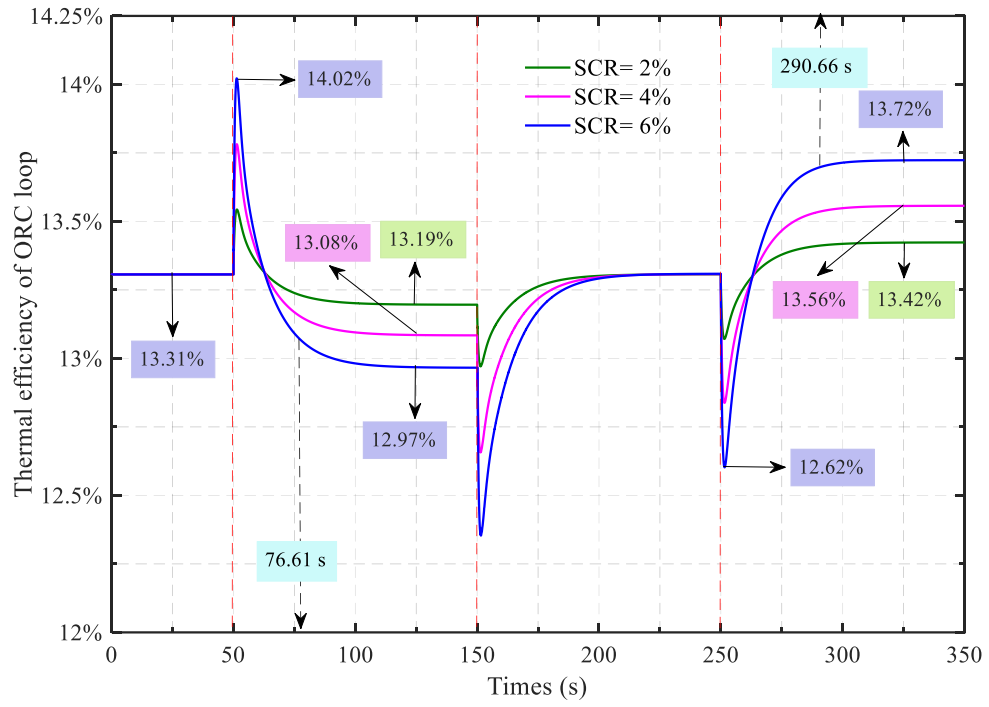
As shown in Fig. 7, when SCR is 6%, 4% and 2%, the average absolute variations of W_{exp} are 1.46 kW, 0.81 kW and 0.43 kW, while that of the W_{net} are 1.09 kW, 0.77 kW and 0.41 kW, respectively. Meanwhile, the average absolute variations of $\eta_{\text{ORC,th}}$ are 0.34%, 0.23% and 0.12%, while that of the $\eta_{\text{ORC-CHP,th}}$ are the least, with 0.13%, 0.08% and 0.02%, respectively. Moreover, it can also be found that the average relative variations of W_{exp} , W_{net} , $\eta_{\text{ORC,th}}$ and $\eta_{\text{ORC-CHP,th}}$ are 4.86%, 4.16%, 2.55% and 0.14%, respectively, with the SCR of 6%. The influence of T_h on W_{exp} is the greatest and that on $\eta_{\text{ORC,th}}$ is the least. The reason for this is that the $\eta_{\text{ORC-CHP,th}}$ is not only affected by the W_{net} , but also affected by the Q_{dhw} , and Q_{dhw} remains stable. In addition, the rise times for W_{exp} , W_{net} , $\eta_{\text{ORC,th}}$ and $\eta_{\text{ORC-CHP,th}}$ are 29.41 s, 28.51 s, 26.61 s and 22.01 s, and the settling times are 42.91s, 42.16s, 40.66s and 42.06s, respectively.



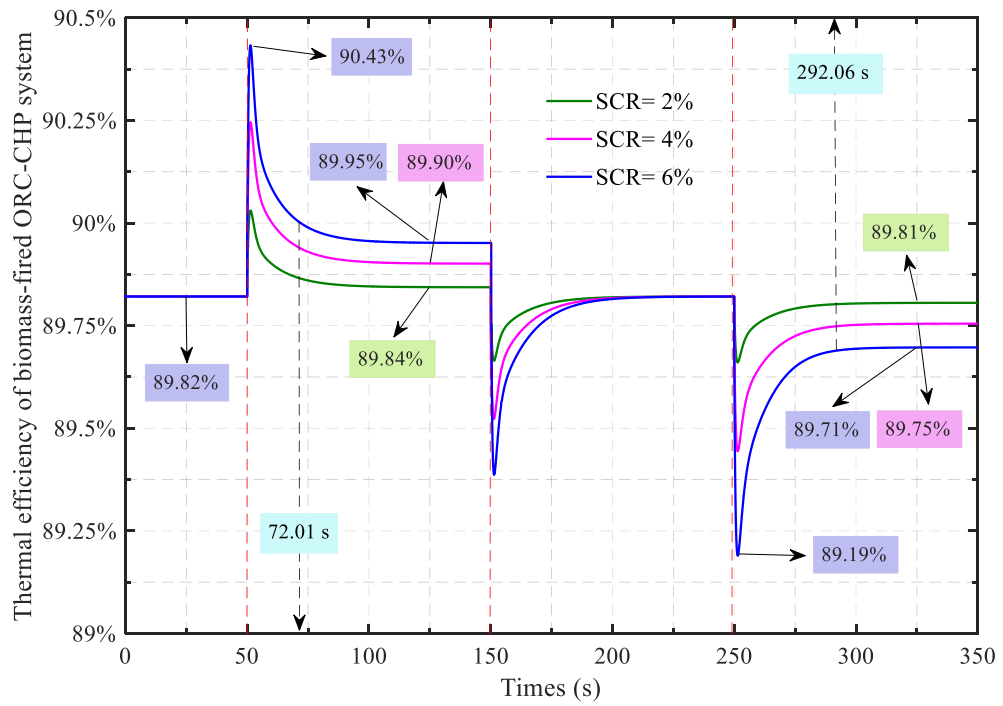
(a)



(b)



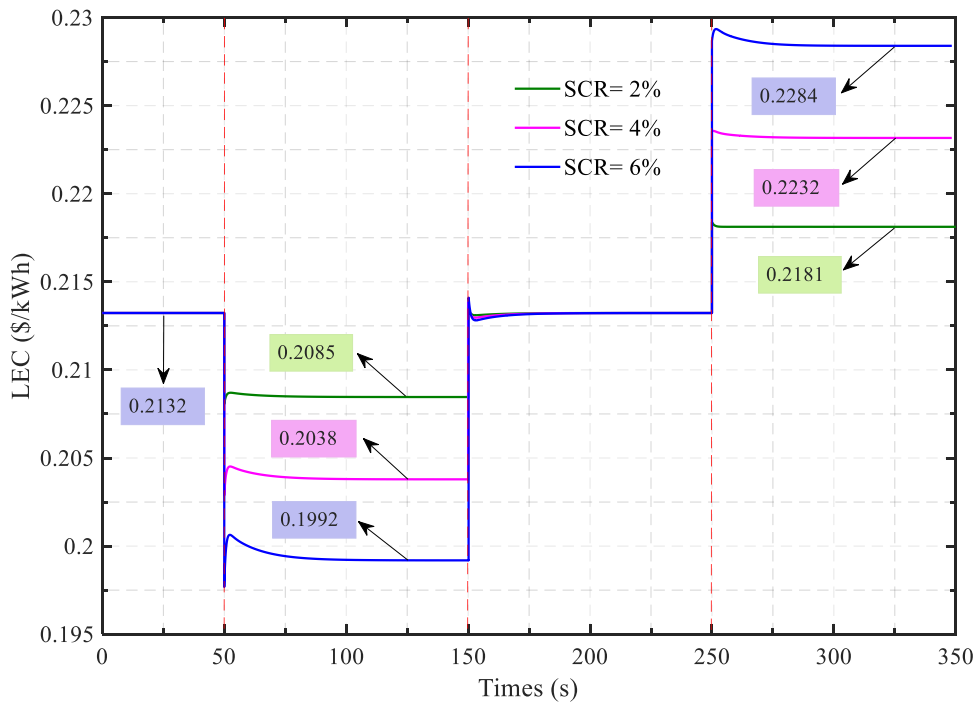
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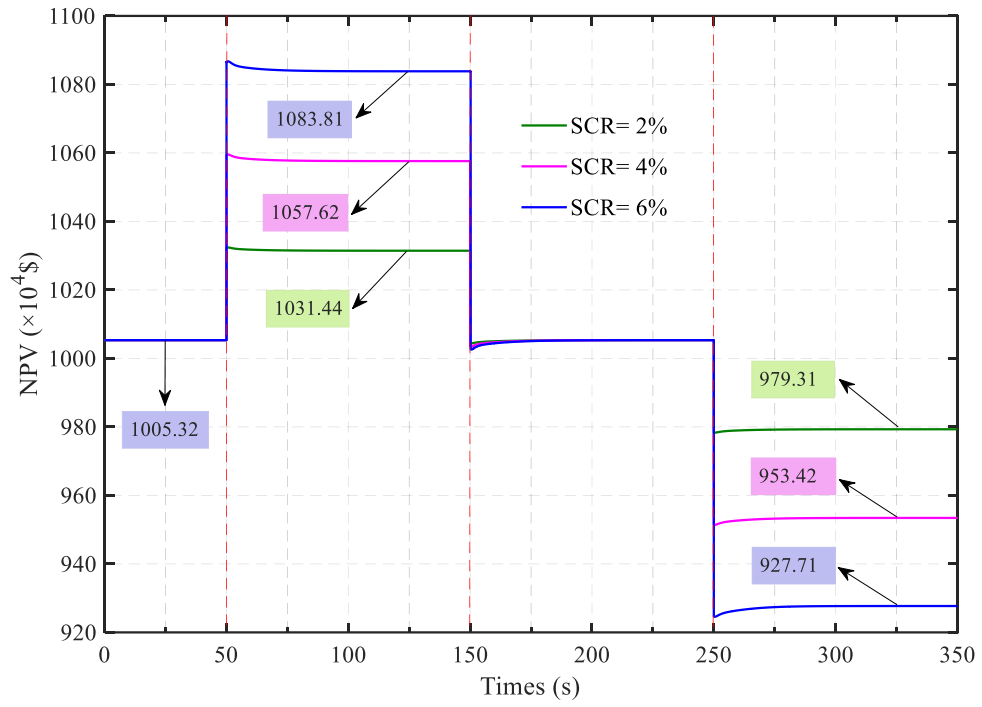
(d)

Fig. 7. Variations of thermodynamic performances with heat source temperature. (a) Expander shaft work. (b) Net output work. (c) Thermal efficiency of ORC loop. (d) Thermal efficiency of biomass-fired ORC-CHP system.

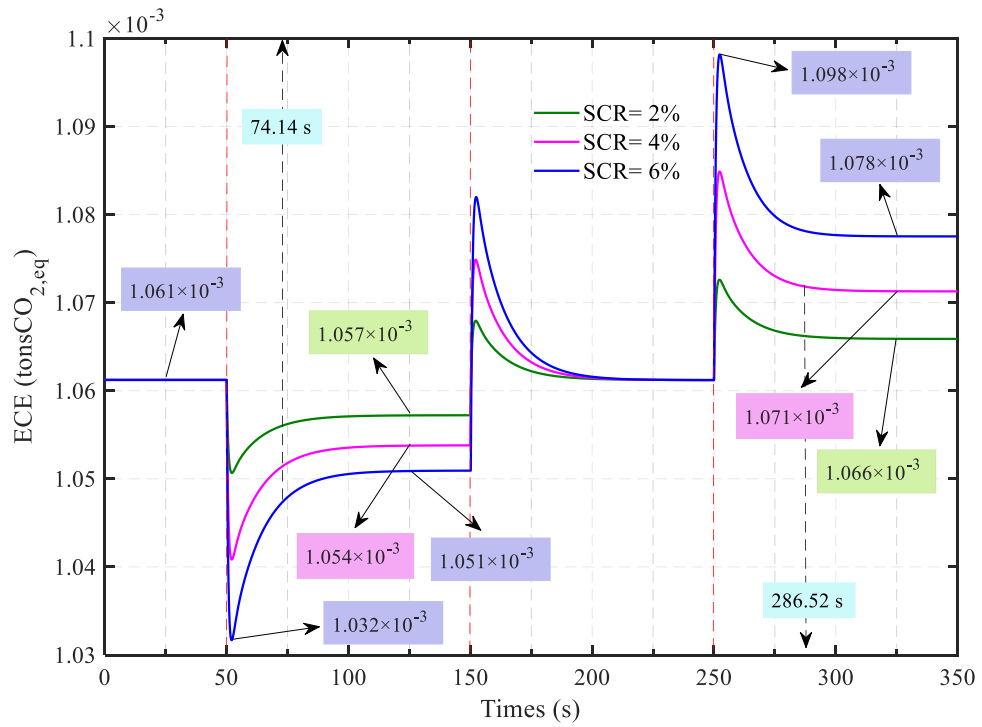
Figure 8 demonstrates the effect of T_h on the dynamic characteristics of the levelized electricity cost (LEC), the net present value (NPV) and Equivalent Carbon Emission (ECE). The LEC and NPV are the economic performance indicators, which can be applied to evaluate the economic feasibility and economic return of the system, respectively. Furthermore, the ECE could help to evaluate the impact on the environment and provide a reference for the sustainable development. The lower LEC and higher NPV represent greater economic performance, and the smaller ECE indicates superior environmental performance. As shown in Fig. 8(a) and Fig. 8(b), with the increase of the T_h , the LEC keeps decreasing, while the NPV displays the opposite trend, implying the better economic performance of the biomass-fired ORC-CHP system. As indicated in Fig. 8(c), the ECE keeps decreasing with T_h , representing a superior environmental performance. Since there are numerous parameters influencing the economic indicators, such as the cost of each major component and biomass fuel, the profit of the expander electricity generation and domestic hot water, so it is difficult to analyse the specific influence mechanism. Furthermore, the rise time and settling time of the ECE are 24.14 s and 36.52 s, respectively.



(a)



(b)



(c)

Fig. 8. Variations of economic and environmental performances with heat source temperature.

(a) *LEC*. (b) *NPV*. (c) *ECE*.

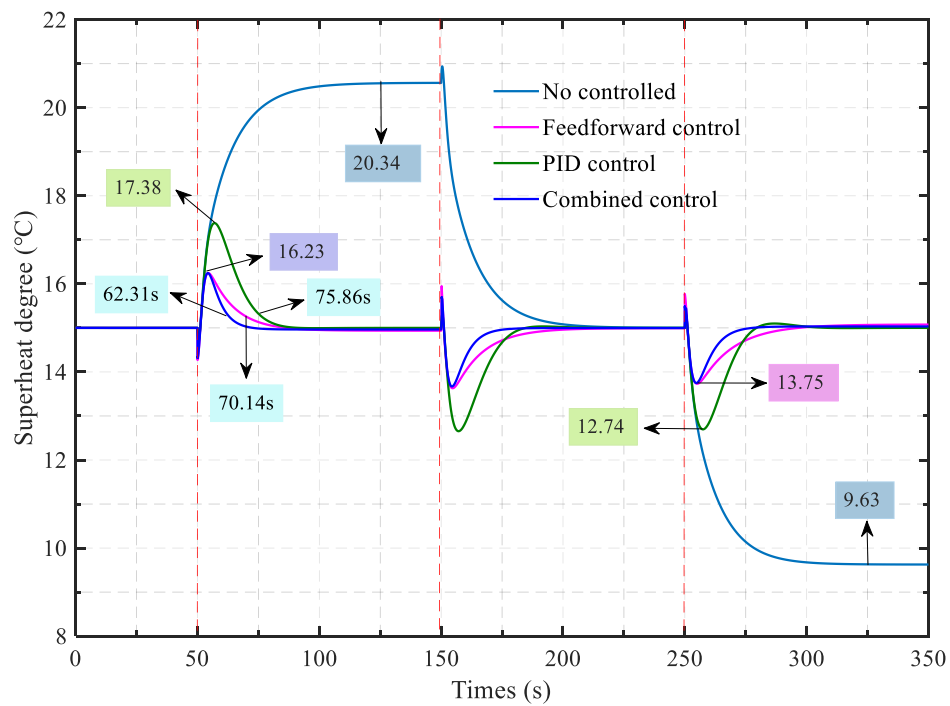
4.2 Effect of controlled superheat on dynamic characteristics

As discussed in section 4.1, it can be found that the fluctuation of heat source has a great influence on the operating parameters of the system, especially for the T_{sup} , where the variation of T_{sup} reaches 7.2 K with the SCR is 6%. However, excessive variations of T_{sup} cause a large impact on the evaporator which might affect the system safety performance. Therefore, three control methods are adopted to reduce the variations of T_{sup} , which ensure the system safety. In this section, the SCR of T_h is selected as 4%, while the characteristics of the heat source are similar to Section 4.1. For the response speed of the control strategy, since the response curve of the T_{sup} under the control strategy presents an underdamped state, the response time is defined as the time to reach the steady state (2% error band is selected in this study).

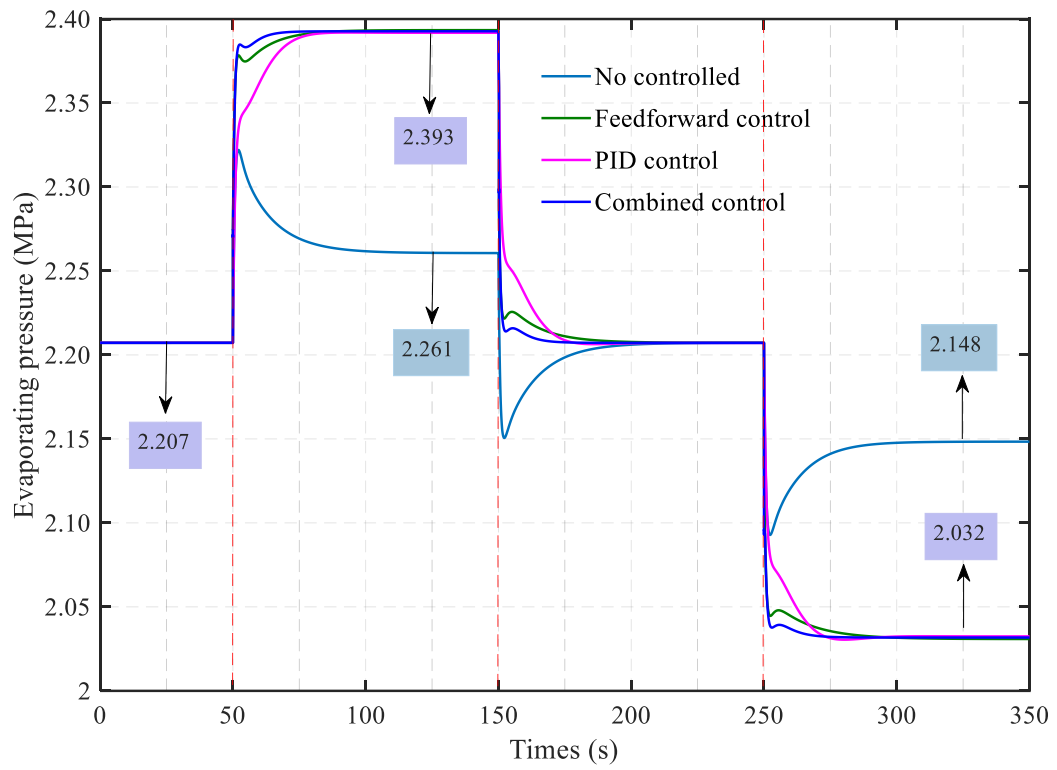
Figure 9 represents the effect of the control strategy of T_{sup} on the dynamic characteristics of T_{sup} , P_{eva} and m_f . In this study, the control method is realized by controlling the pump speed and hence changing the m_f . As shown in Fig. 9(a), it is clear that the three control methods have excellent control effect on T_{sup} . The T_{sup} displays a parabolic trend, with the overshoot of 2.38 K for the PID control, the feedforward control and the combined control exhibit the same overshoot of 1.23 K during 50-150 s. From Fig. 9(b) and Fig. 9(c), it can be found that the control strategy has a significant effect on the final values of P_{eva} and m_f , but the final values of the three control methods are the same. The absolute variations of P_{eva} and m_f with no control are only 53.5 kPa and 0.0023 kg/s respectively, but under the control strategy of T_{sup} , the absolute variations of P_{eva} and m_f increase to 186.5 kPa and 0.03745 kg/s respectively. Compared to without control, the absolute variation of P_{eva} and m_f is increased by 248% and 1485% under the control strategy. These findings can be explained that the three control methods exploit altering the pump speed and then varying the m_f to achieve the control effect. Therefore, in order to maintain the stable of T_{sup} , it is necessary to enhance the m_f , causing the P_{eva} also increases, but the increase rate of P_{eva} is much smaller than the m_f .

For the three controls methods, it can be found that the PID control has the longest response time of 25.86 s, the combined control is the shortest of 12.31 s, and the feedforward control is 20.14 s. Compared to without control, the dynamic response times of PID control, feedforward control and

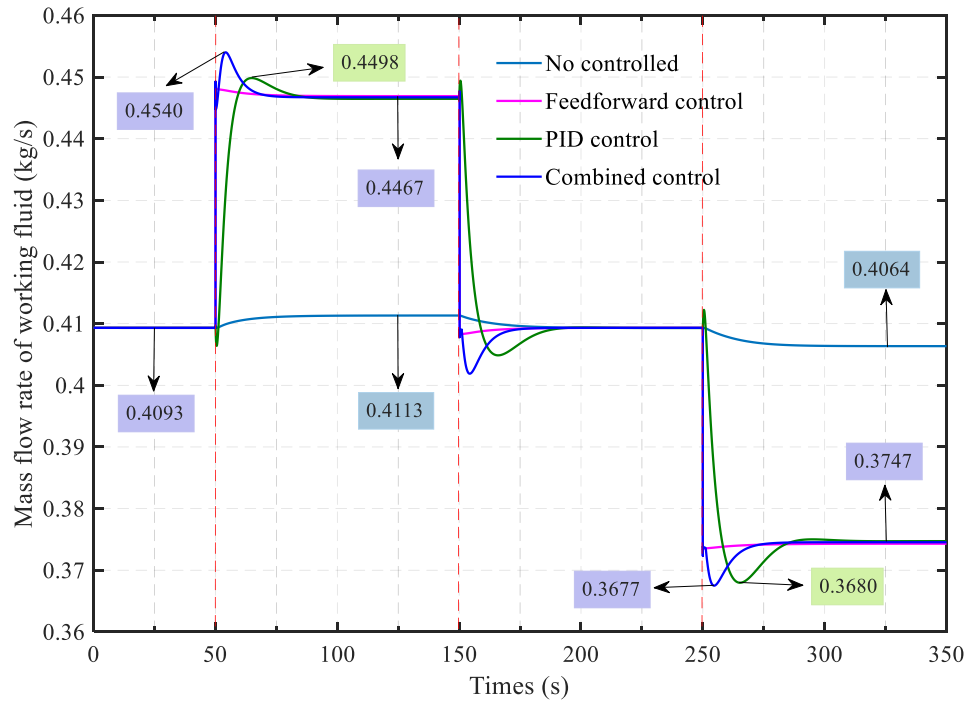
combined control improved by -7.12%, 16.57% and 49.01%, respectively. This is because PID control is a passive correction based on the deviation of real-time operating data from the target value, while feedforward control is an active correction based on the heat balance, so the response speed of feedforward control is faster than PID control. For the combined control, it is mainly driven by the feedforward control, and the PID control provides secondary correction, resulting in the fastest control response time.



(a)



(b)



(c)

Fig. 9. Variations of key operating parameters with heat source temperature under the control strategy of superheat degree. (a) Superheat degree of evaporator outlet. (b) Evaporating pressure. (c)

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Mass flow rate.

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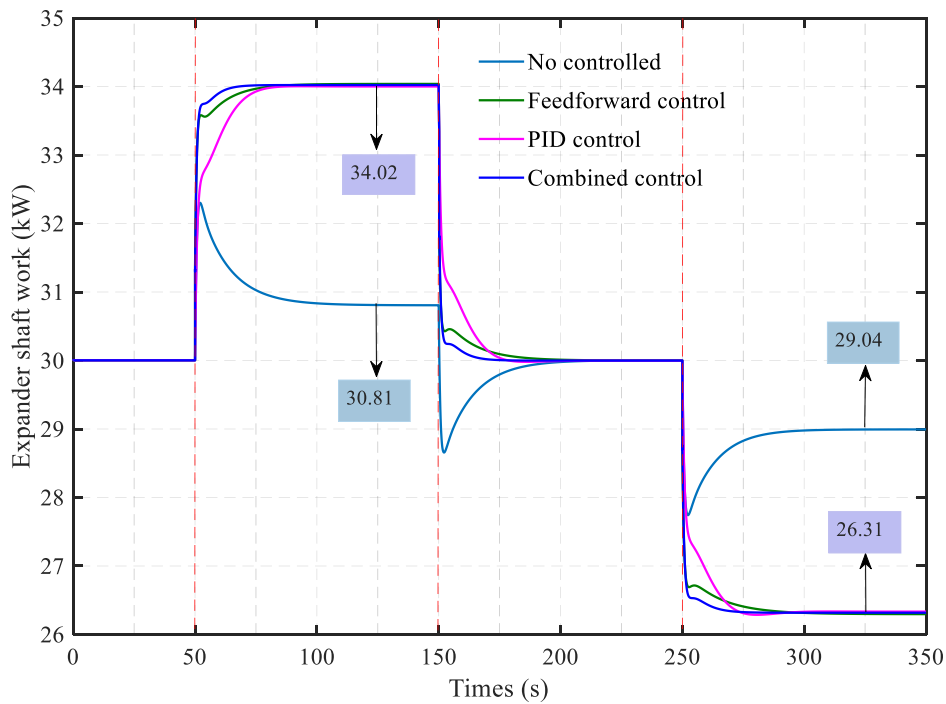
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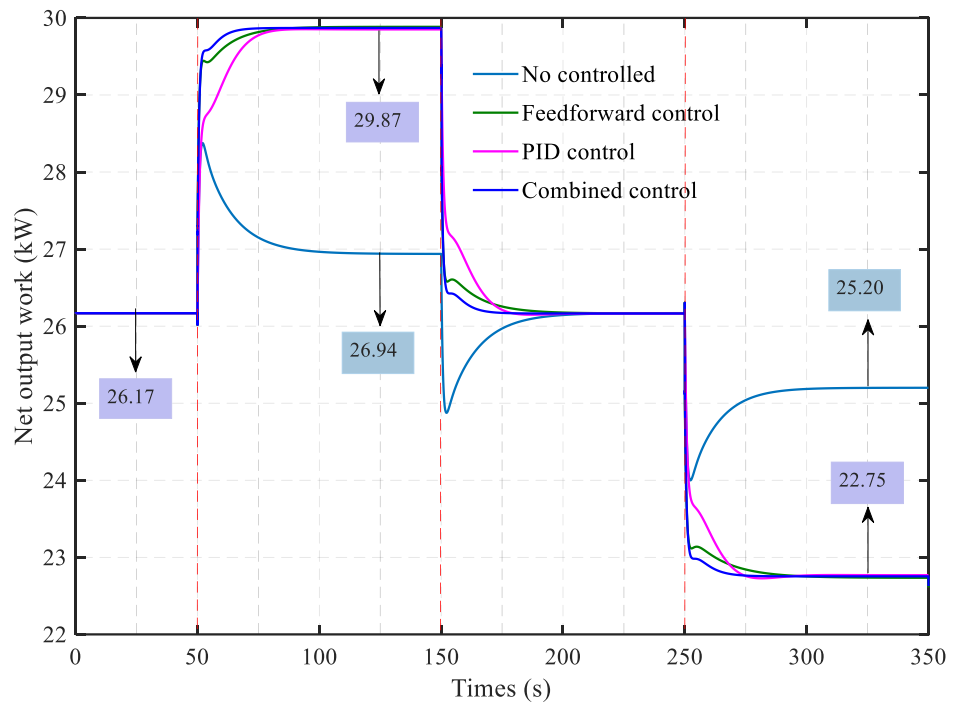
The effect of the control strategy of T_{sup} on the dynamic characteristics of W_{exp} , W_{net} , $\eta_{\text{ORC,th}}$ and $\eta_{\text{ORC-CHP,th}}$ is shown in Fig. 10. As seen in Fig. 10(a) and Fig. 10(b), compared to without control, the increase in m_f leads to an increase in the variation of W_{exp} and W_{net} . The absolute variation of the W_{exp} and W_{net} without control are 0.81 kW and 0.77 kW, respectively, whereas that with the control strategy are 4.02 kW and 3.7 kW, which increase by 396% and 381%, respectively. From Fig. 10(c), It should be noted that the $\eta_{\text{ORC,th}}$ exhibits an increasing trend with T_h under the control strategies, presenting the opposite trend of without control. The reason for this is that the W_{net} appears greater enhancement under the control effect, while the Q_{eva1} increases smaller, leading to an increase in $\eta_{\text{ORC,th}}$. It can be seen from Fig. 10(d) that the effect of the control strategy on the $\eta_{\text{ORC-CHP,th}}$ is small. The reason for this is that the $\eta_{\text{ORC-CHP,th}}$ is affected by the W_{net} and the Q_{dhw} , while the variation of m_f has little effect on the Q_{dhw} . Therefore, considering the thermodynamic performance of the biomass-fired ORC-CHP system, the control strategy of T_{sup} has a significant improvement when the T_h keeps increasing.



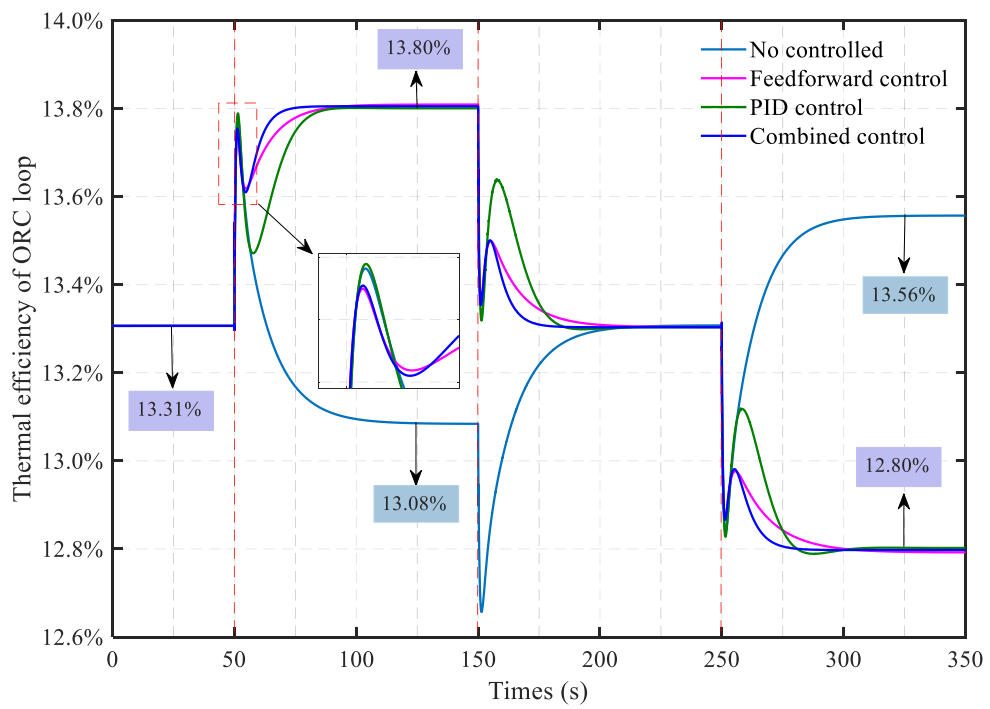
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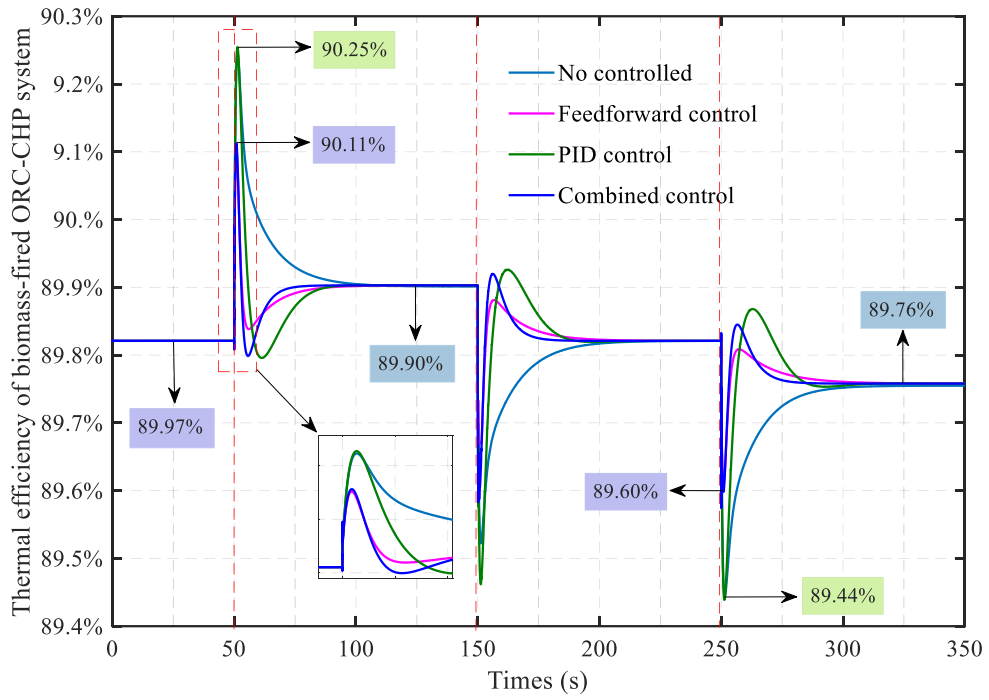
(a)



(b)



(c)



(d)

Fig. 10. Variations of thermodynamic characteristics with heat source temperature under the control strategy of superheat degree. (a) Expander shaft work. (b) Net output work. (c) Thermal efficiency of ORC loop. (d) Thermal efficiency of biomass-fired ORC-CHP system.

Table 9 compares the overshoots and response times for different control methods under the control strategy of T_{sup} . In terms of different control methods, PID control is the most classical model-free closed-loop control method, which can be widely implemented in various control strategies. However, it can be seen from Table 9 that the control response time is long with 25.86 s. For systems requiring fast response and high accuracy, a further optimization or integration with additional control methods may be necessary. The feedforward control has a significantly faster control response time of 20.14 s, which may be contributed to the need of calculating the control quantities in advance and acting directly on the control object. However, the variation of heat source temperature is random and irregular in practical applications, and how to calculate the control quantity according to the random heat source temperature and improve the accuracy of the calculation model become the focus of future research. The combined control deserves more attention as a novel control method in more demanding control strategies, which not only utilizes feedforward control to improve the dynamic response

performance, but also provides steady state error compensation through PID control.

For the control strategy of T_{sup} , although the primary purpose is to guarantee the security of the biomass-fired ORC-CHP system, it is also desired to improve the system performance on this basis. From the Table 9, it can be seen that this requirement is fulfilled when the T_h increases, in which the output power and system efficiency are enhanced. In order to minimize the disturbance to the system caused by the rise in T_h , a higher m_f is required to reduce the heat transfer rate of per unit mass flow rate, thereby mitigating this effect.

Table 9. Comparison of three control methods in the control strategy of superheat degree

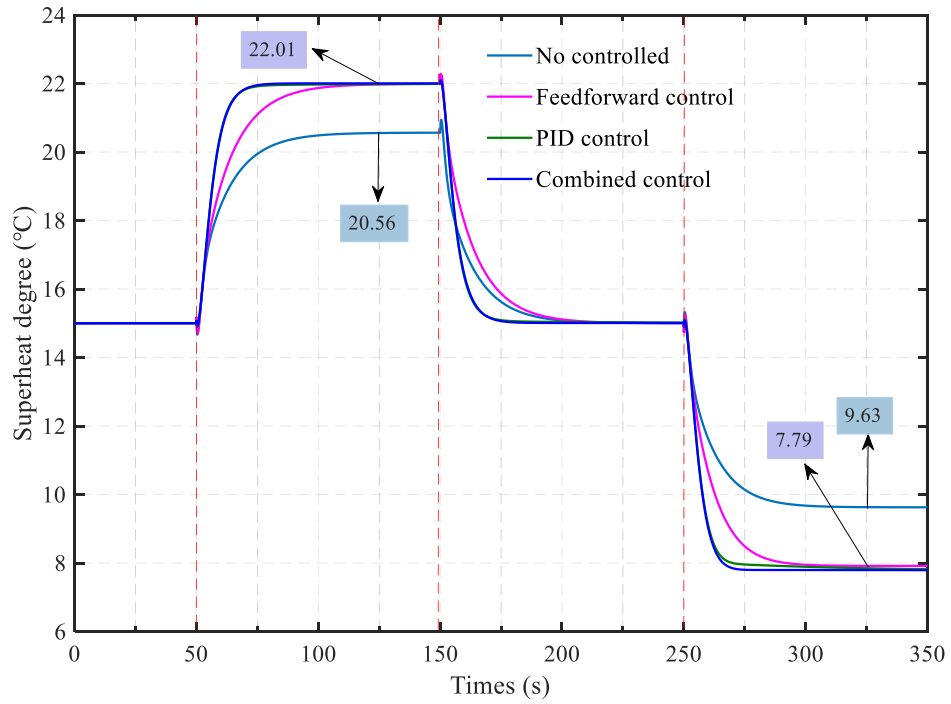
	Feedforward controlled		PID controlled		Combined controlled	
	Overshoot	Response time	Overshoot	Response time	Overshoot	Response time
T_{sup} (°C)	1.23	20.14 s	2.38	25.86 s	1.23	12.31 s
P_{eva} (MPa)	0.186	-	0.186	-	0.186	-
m_f (kg/s)	0.0374	0.76 s	0.0447	15.18 s	0.0405	9.35 s
W_{exp} (kW)	4.02	-	4.02	-	4.02	-
W_{net} (kW)	3.7	-	3.7	-	3.7	-
$\eta_{\text{ORC,th}}$	0.49%	-	0.49%	-	0.49%	-
$\eta_{\text{ORC-CHP,th}}$	0.14%	-	0.28%	-	0.14%	-

4.3 Effect of controlled expander shaft work on dynamic characteristics

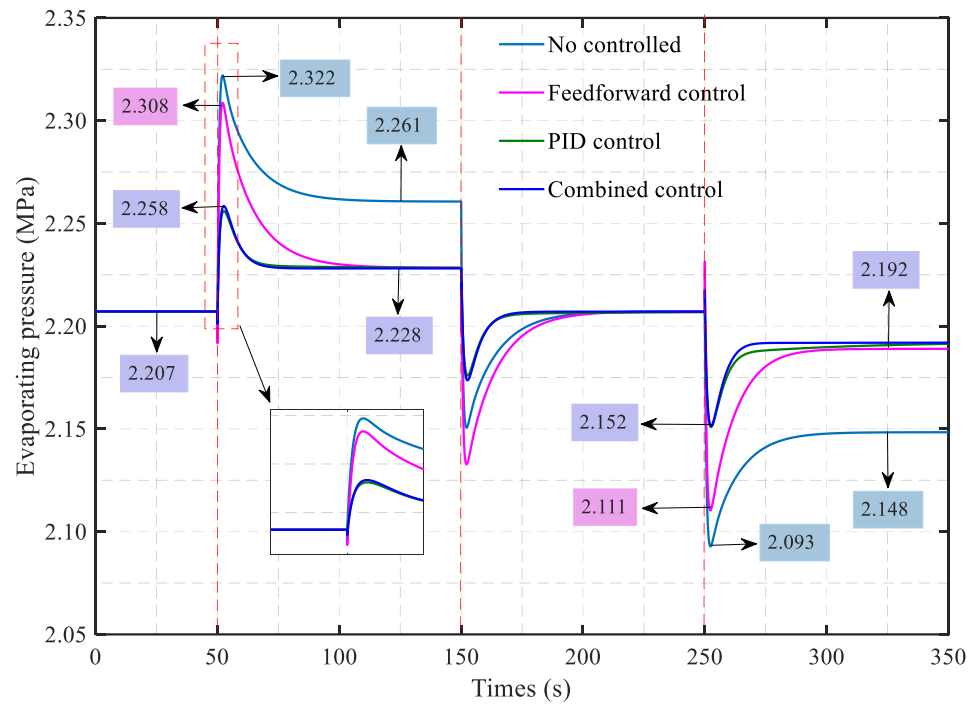
As demonstrated in Section 4.2, the three control methods present better control effects in the T_{sup} , but have a strong impact on the W_{exp} , resulting in the instability of the W_{net} . However, the unstable of output work will cause a serious damage to the generator, and it is expected to obtain a stable output work in most practical situations. Therefore, in this section, the W_{exp} is selected as the control target and three control methods are adopted to ensure the system stability. Similar to the control strategy of T_{sup} , the SCR of T_h is 4%, and the variation time is 350 s.

Figure 11 represents the effect of the control strategy for W_{exp} on the dynamic characteristics of T_{sup} , P_{eva} and m_f . Unlike the control strategy of T_{sup} , the T_{sup} exhibits an increasing trend compared to without control with the increase of T_h , while the P_{eva} and m_f decrease. The absolute variation of T_{sup}

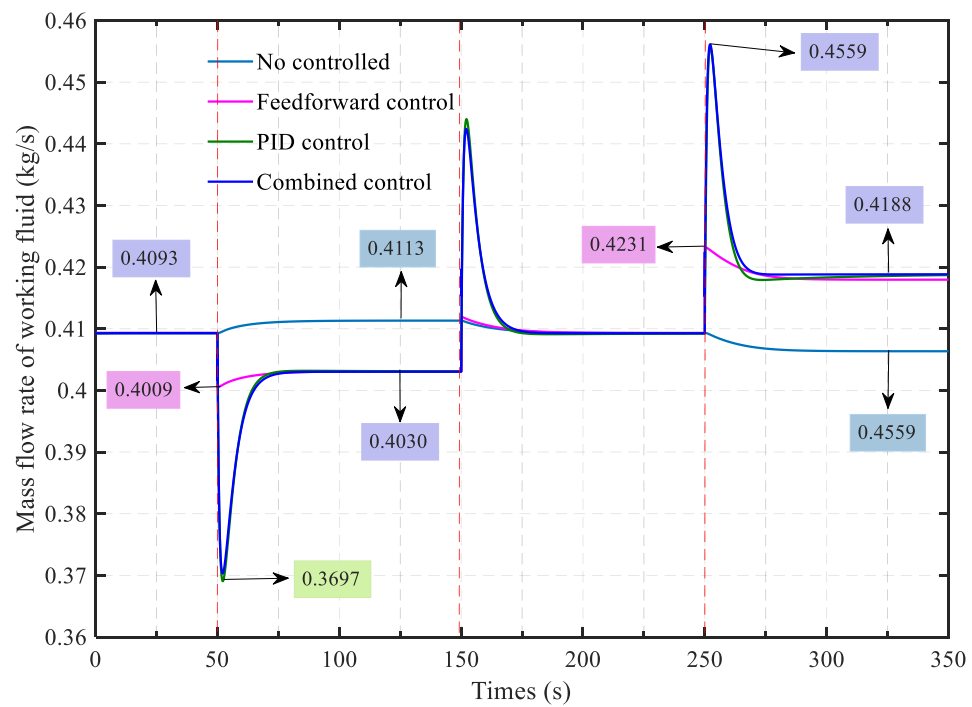
under the control strategy is 7.01 K, which increases by 26.08% compared to without control, and decreases by 61.11% of P_{eva} . In this control strategy, it is necessary to achieve the control effect of the W_{exp} by reducing the m_f when the T_h rises, and hence the m_f exhibits a decreasing trend compared to without control. As shown in Fig. 11(c), it can be seen that the steady state of m_f decreases by 0.0063 kg/s from 50 to 150 s, and decreases by 0.0083 kg/s compared to without control.



(a)



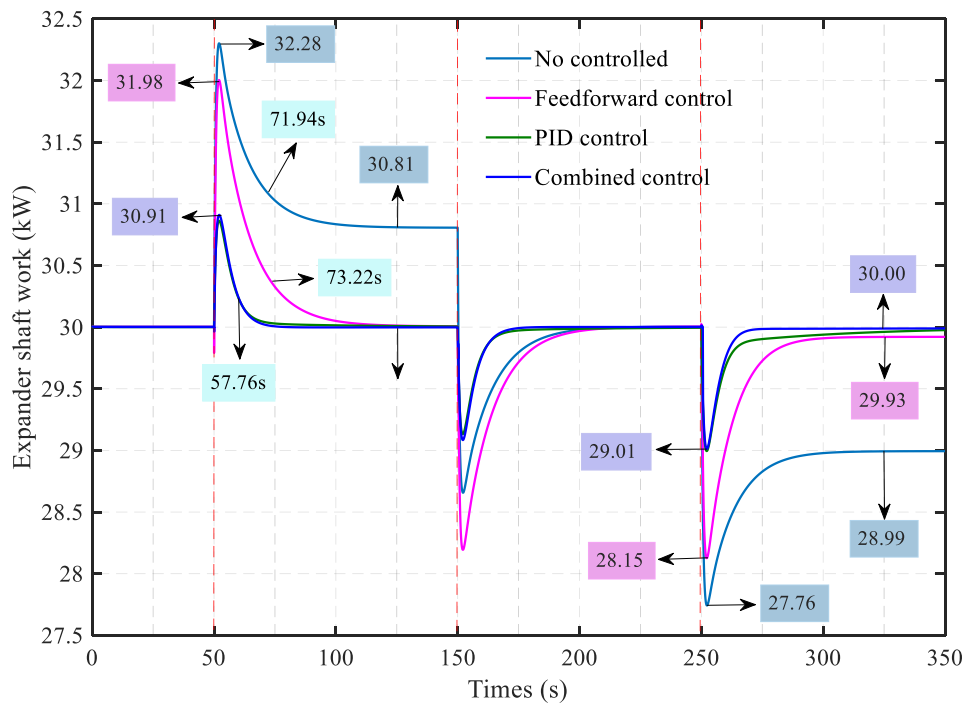
(b)



(c)

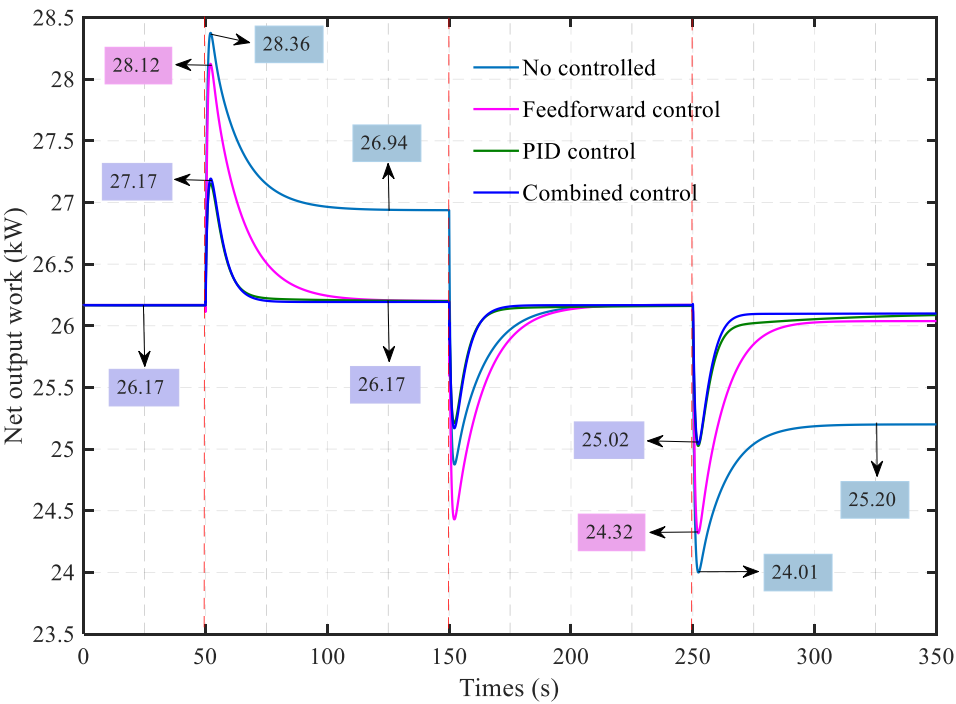
Fig. 11. Variations of key operating parameters with heat source temperature under the control strategy of expander shaft work. (a) Superheat degree of evaporator outlet. (b) Evaporating pressure. (c) Mass flow rate.

Figure 12 represents the effect of the control strategy for W_{exp} on the dynamic characteristics of W_{exp} , W_{net} , $\eta_{\text{ORC,th}}$ and $\eta_{\text{ORC-CHP,th}}$. As shown in Fig. 12(a), the three control methods exhibit an excellent control effect in W_{exp} , and the PID control and combined control have the same overshoot and response time, which are 0.91 kW and 7.76 s, respectively. However, it can also be found that the combined control reaches stabilization faster compared to the PID control. The reason for this is that the combined control is also influenced by the feedforward control, resulting in the increase in the response speed. Moreover, the feedforward control shows the worst control effects, while the overshoot and response time are 1.98 kW and 21.94 s, respectively. Similar to the W_{exp} , the combined control method also has the best control effect on the W_{net} , and the feedforward control is the worst. From Fig. 12(c) and Fig. 12(d), since the W_{exp} remains 30 kW under the control strategy, the Q_{eval} keeps increasing at 50-150 s, resulting in a decrease in the $\eta_{\text{ORC,th}}$. The effect of the control strategy on the $\eta_{\text{ORC-CHP,th}}$ is minor and mainly receives the influence of T_h . In contrast to the control strategy for T_{sup} , more emphasis should be placed on the phase of T_h decrease occurring between 250-350 s. It can be found with the decrease of the T_h , the W_{exp} , W_{net} and $\eta_{\text{ORC,th}}$ increase by 0.94 kW, 0.96 kW and 0.32%, respectively. The control strategy of W_{exp} significantly enhances the thermodynamic performance of biomass-fired ORC-CHP system when the T_h decreases.



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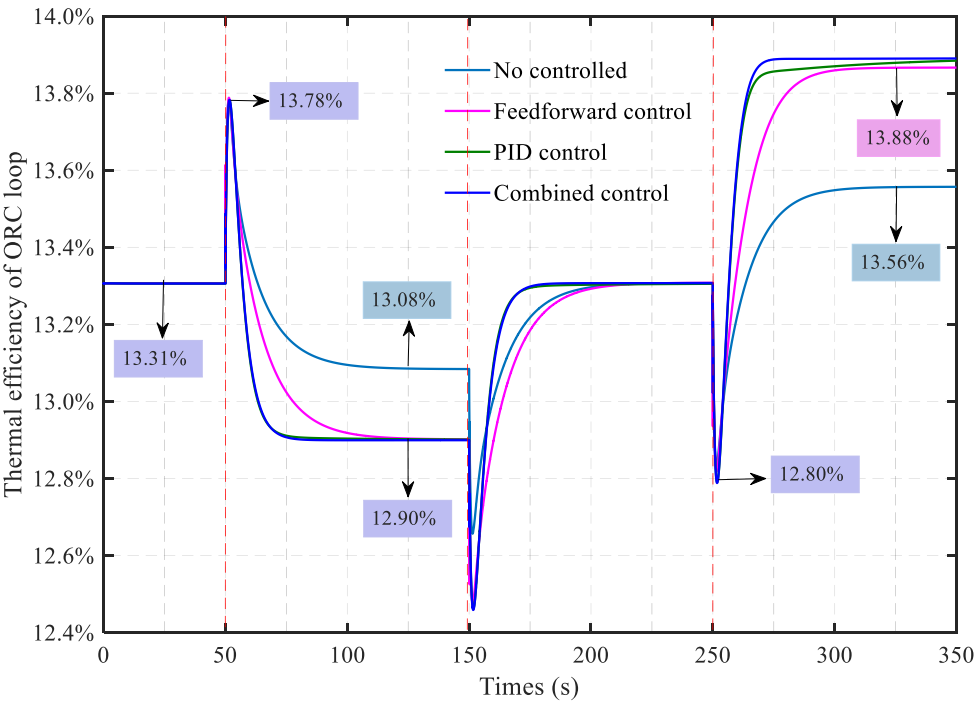
(a)



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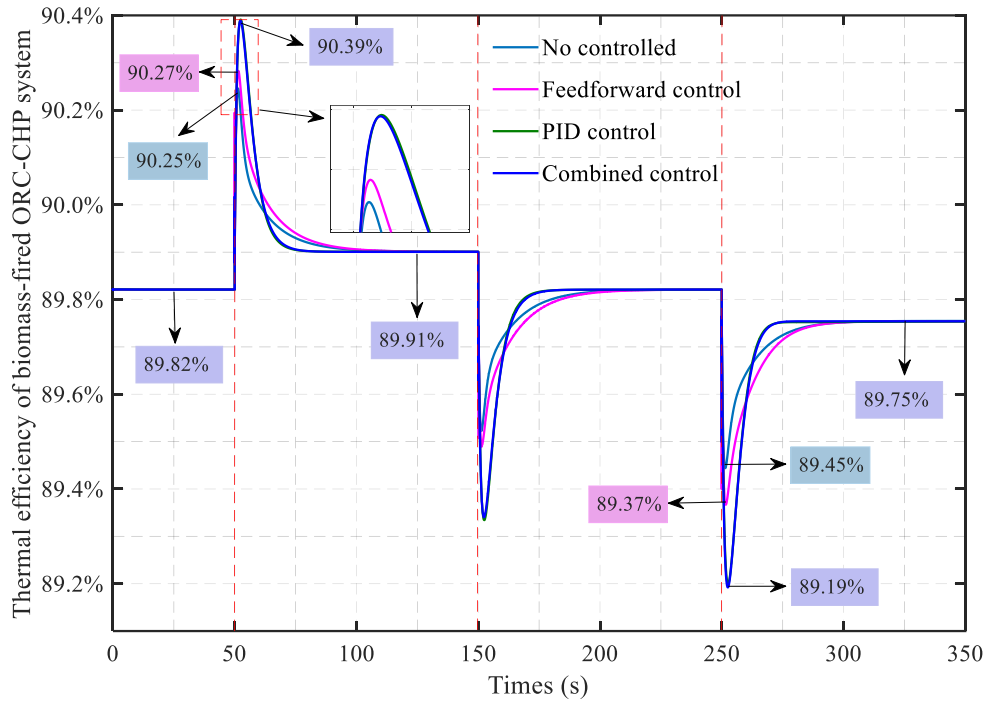
(b)



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(c)



(d)

Fig. 12. Variations of thermodynamic characteristics with heat source temperature under the control strategy of expander shaft work. (a) Expander shaft work. (b) Net output work. (c) Thermal efficiency of ORC loop. (d) Thermal efficiency of biomass-fired ORC-CHP system.

Table 10 compares the overshoots and response times for different control methods under the control strategy of W_{exp} . Different from the control strategy of T_{sup} , the control strategy of W_{exp} is more complex due to it is determined by more parameters, which including m_f , inlet and outlet pressures of the expander. The feedforward control method demonstrates inadequate control behaviour compared to more complex strategies, as evidenced by the longest response time recorded in the table at 23.22 s. This is because feedforward control requires considerations of the effect of m_f on the inlet and outlet expander pressures when altering the m_f , which requires a higher accuracy of the model compared to PID control, thereby restricting the application of feedforward control in complex control strategies. PID control shows relatively strong stability in various control situations, but the workload of parameter adjustment will increase for complex control strategies. Combined control still performs the best control effect, while the overshoot and settling time are 0.99 kW and 17.76 s, respectively. For the control strategy of W_{exp} , the m_f is increased under the control strategy to

ensure the stability of the W_{exp} when the T_h decreases. The system performance improved with this strategy, whereas the T_{sup} at the evaporator outlet decreases. Therefore, achieving a balance between these factors becomes crucial in practical applications.

Table 10. Comparison of three control methods in the control strategy of expander shaft work

	Feedforward controlled		PID controlled		Combined controlled	
	Overshoot	Response time	Overshoot	Response time	Overshoot	Response time
T_{sup} (°C)	-7.21	-	-7.21	-	-7.21	-
P_{eva} (MPa)	-0.096	-	-0.055	-	-0.055	-
m_f (kg/s)	0.0138	23.22 s	0.0466	21.94 s	0.0466	17.76 s
W_{exp} (kW)	-1.85	23.22 s	-2.24	21.94 s	-0.99	17.76 s
W_{net} (kW)	-1.85	-	-2.16	-	-1.15	-
$\eta_{\text{ORC,th}}$	0.57%	-	0.57%	-	0.57%	-
$\eta_{\text{ORC-CHP,th}}$	-0.45%	-	-0.37%	-	-0.63%	-

The variations of LEC , NPV and ECE under different control strategies are shown in Fig. 13. From Fig 13, it can be found that the control strategy of T_{sup} has a greater impact on the economic performance of the biomass-fired ORC-CHP system compared to the control strategy of W_{exp} . The LEC decreases by 12.4×10^{-5} \$/kWh and 5.1×10^{-5} \$/kWh under the control strategy of T_{sup} and W_{exp} compared to without control, while the NPV increases by 9.4×10^{-3} \$ and 9.3×10^{-3} \$, respectively. Furthermore, it also can be found that the environmental performance is better under the control strategy of T_{sup} , while the variation of ECE reaches 3.24×10^{-5} tonsCO_{2,eq}. In conclusion, the control strategy of T_{sup} presents a relatively higher economic and environmental performance of the biomass-fired ORC-CHP system compared to the control strategy of W_{exp} . The reason for this is that the control strategy of W_{exp} limits the output power of the system, resulting in the reduction of economic and environmental performance.

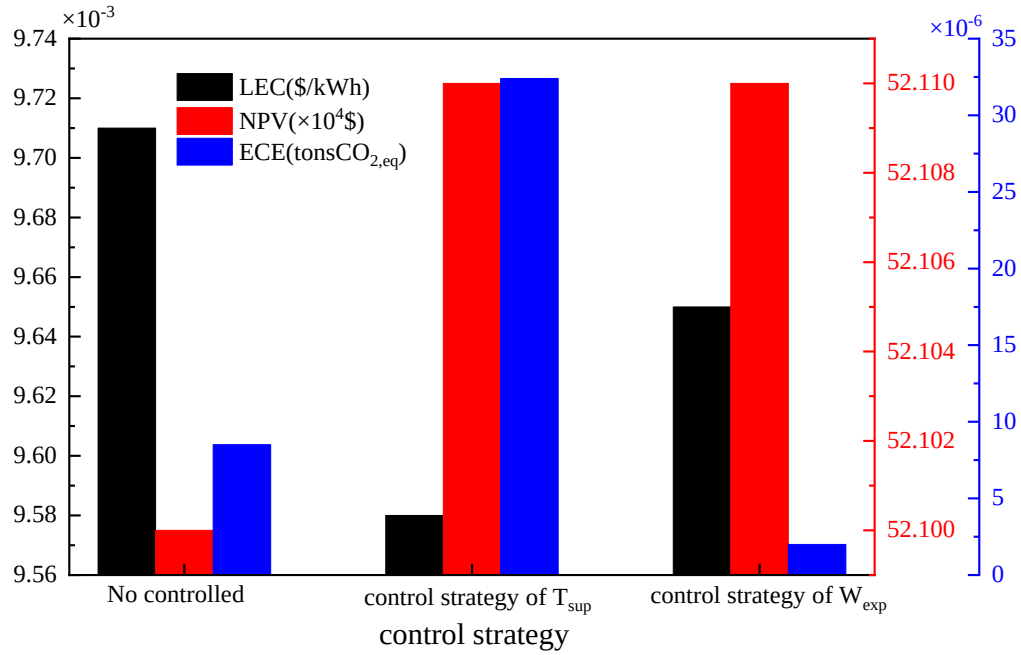


Fig. 13. Variations of LEC , NPV and ECE under different control strategies.

5. Conclusion

The dynamic behaviour and performance evaluation of a biomass-fired ORC-CHP system under different control strategies are investigated. The dynamic model of a biomass-fired ORC-CHP system is developed using Simulink software. The dynamic behaviours of the key operating parameters (T_{sup} , P_{eva} and m_f) and thermodynamic, economic and environmental performance with the heat source temperature are examined. Furthermore, in order to maintain the safety and the stability of the output power, two control strategies (control strategy of T_{sup} and W_{exp}) and three control methods (feedforward control, PID control and combined control) are applied in this study. The conclusions obtained are as follows:

(1) In terms to the dynamic behaviour of key operating parameters, the T_h has the greatest effect on T_{sup} with the variation of 8.36 K when the SCR is 6%, while the m_f is the least of 0.0029 kg/s. Furthermore, the T_{sup} presents the fastest response speed with the rise time of 21.14 s. The $\eta_{ORC,th}$ shows a decreasing trend with T_h , while the W_{exp} increases by 4.02 kW when the SCR is 6%.

(2) The combined control performs the best control effect in the control strategies of T_{sup} and

W_{exp} , while the control response time are 12.31 s and 7.76 s, respectively. Moreover, the feedforward control method exhibits a poor control behaviour under the control strategy of W_{exp} , while PID control displays a relatively strong stability in various control situations.

(3) The control strategy of T_{sup} enhances the system performance when the T_{h} increases, whereas that of W_{exp} exhibits a system improvement when the T_{h} decreases. Compared to without control, the W_{exp} and $\eta_{\text{ORC,th}}$ of the control strategy of T_{sup} increase by 3.21 kW and 0.78%, while that of the control strategy of W_{exp} increase by 1.01 kW and 0.32%, respectively.

(4) The control strategy of T_{sup} presents a significant improvement in the economic and environmental performance of the biomass-fired ORC-CHP system.

This study focuses on the dynamic behaviour of biomass-fired ORC-CHP system under different control methods and control strategies, and the results can provide theoretical references for industrial applications. However, the response speed of PID control is weak and feedforward control performs poorly under complex control strategies. How to improve the response speed and accuracy of the control method and how to de-balance the relationship between T_{sup} and W_{exp} under the control strategy are expected to be the focus of subsequent research.

Acknowledgements

This research work has been supported by the Key Research and Development Program of Jiangsu Province, China (BE2019009-4), the National Natural Science Foundation of China (51806081), the Key Research and Development Program of Taizhou City, China (TS202201), the 2019 Scholarship of the Knowledge Center on Organic Rankine Cycle (KCORC, www.kcorc.org) and the Open Foundation Program of Key Laboratory of Wind Energy and Solar Energy Technology (Inner Mongolia University of Technology), Ministry of Education of China (2020ZD02). The authors gratefully acknowledge financial support from China Scholarship Council (CSC).

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