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An effective thermodynamic transformation analysis method for actual irreversible cycle

CHEN ZeShao[1], XIE WenHai[1], HU Peng[1*], JIA Lei[2] & SHI Min[2]

[1] *Department of Thermal Science & Energy Engineering, University of Science and Technology of China, Hefei 230027, China;*
[2] *State Key Laboratory of Compressor Technology, Hefei Institute of General Machinery, Hefei 230027, China*

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An effective thermodynamic transformation analysis method was proposed in this study. According to the phenomenon of exergy consumption always coupling with heat transfer process, the effective thermodynamic temperatures were defined, then the actual power cycle or refrigeration/heat pump cycle was transformed into the equivalent reversible Carnot or reverse Carnot cycles for thermodynamic analysis. The derived effective thermodynamic temperature of the hot reservoir of the equivalent reverse Carnot cycle is the basis of the proposed method. The combined diagram of TR-h and TR-q was adopted for the analysis of the system performance and the exergy consumption, which takes advantage of the visual expression of the heat/work exchange and the enthalpy change, and is convenient for the calculation of the coefficient of performance and exergy consumptions. Take a heat pump water heater with refrigerant of R22 for example, the proposed method was systematically introduced, and the fitting formulas of the effective thermodynamic temperatures were given as demonstration. The results show that the proposed method has advantage and well application foreground in the performance simulation and estimation under the variable working conditions.

irreversible thermodynamic cycle, effective thermodynamic transformation analysis, effective thermodynamic temperature, heat pump water heater, exergy analysis

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1 Introduction

The actual thermodynamic/refrigeration/heat pump system consists of the cycle of the working fluid and the heat exchange between the working fluid and the thermal reservoirs, which make unavoidable exergy consumption. How to express the relationship of the energy transfer, conversion, and the exergy consumption is the joint theme of thermodynamics and heat transfer. The classical thermodynamics [1, 2] adopts the equilibrium state and the reversible processes to study the conversion of heat and work, and the energy conversion efficiency is an ideal reference result. The performance of the actual system has to be obtained from the modification of the ideal result. The exergy analysis method [3, 4] or entropy generation method [5] based on the second law of thermodynamics can analyze the exergy or entropy generation of the processes of a given system with known parameters and give a qualitative guide to the performance improvement. However, the optimization relations of heat transfer, work output, and exergy consumption are still not clear. Finite time thermodynamics [6] can cover this shortage to a certain extent by introducing time parameter. Though great progress has been made [7–9], it still has a certain gap to the actual system which has the effect of the finite thermal capacity of thermal reservoirs, heat transfer,

*Corresponding author (email: hupeng@ustc.edu.cn)

irreversible processes, and real working fluids. Therefore, an effective thermodynamic transformation analysis method was proposed in this study. By taking a heat pump water heater for example, the actual irreversible cycle was transformed into the equivalent reversible cycle for thermodynamic analysis.

2 Heat pump water heater system

Air-source heat pump water heater has three systems: refrigerant, water and air. The cycle of refrigerant is shown in Figure 1. The ideal cycle 12341 consists of four processes: isentropic compression 12, isobaric condensation 23, isenthalpic throttling 34, and isobaric evaporation 41.1″ and 2″ are the suction and exhaust state points of actual compress process which has irreversible loss. The energy exchanges between refrigerant and water/air with variable temperature are shown in Figure 2. For ideal cycle 1234, according to the first law of thermodynamics, there are

$$q_h = h_2 - h_3, \tag{1}$$

$$w = h_2 - h_1, \tag{2}$$

$$q_e = h_1 - h_4, \tag{3}$$

$$COP = q_h / w, \tag{4}$$

where q_h is the theoretical heat dissipation; q_e is the specific cooling capacity (or heat absorption of the evaporator); w is the specific power input, the cooling capacity q_e; h_1, h_2, h_3 and h_4 are the specific enthalpies of refrigerant at corresponding states.

3 Effective thermodynamic temperatures

During thermodynamic process, there may exist heat or work exchange between the system and the surroundings, accompanied by the transfer and consumption of exergy. To describe the process exergy change conveniently, a new parameter named effective thermodynamic temperature was defined in this study.

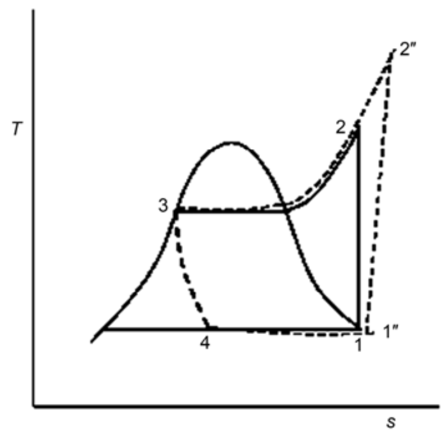

Figure 1 *T-s* diagram for heat pump water heater cycle.

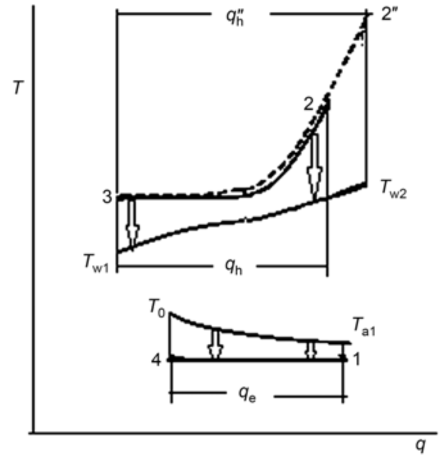

Figure 2 *T-q* diagram for heat pump water heater cycle.

3.1 Effective thermodynamic temperature of reversible process

According to the relationship of entropy change, thermal energy change and temperature in a reversible process $\delta q = T\mathrm{d}s$, we define an effective thermodynamic temperature $T_{R,ab}$ as the thermal energy change to the entropy change during a reversible process:

$$T_{R,ab} = \frac{q_{ab}}{\Delta s_{ab}} = \frac{h_a - h_b}{s_a - s_b}, \tag{5}$$

where h is the specific enthalpy and s is the specific entropy of working fluid. Subscripts a and b are the starting point and the ending point of a thermodynamic process (ab). According to eq. (5), $T_{R,23}$ of isobaric condensation process in Figure 1 is

$$T_{R,23} = \frac{h_2 - h_3}{s_2 - s_3} = \frac{q_h}{\Delta s_{23}}. \tag{6}$$

$T_{R,14}$ of isobaric evaporation process, which is also the temperature of cold reservoir $T_{R,e}$ of equivalent reverse Carnot cycle, is defined as

$$T_{R,e} = T_{R,14} = \frac{h_1 - h_4}{s_1 - s_4} = \frac{q_e}{\Delta s_{14}}. \tag{7}$$

3.2 $T_{R,h}$ of effective reverse Carnot cycle

The core of the effective thermodynamic transformation analysis method is how to obtain the effective thermodynamic temperature of hot reservoir $T_{R,h}$ of equivalent reverse Carnot cycle. For this purpose, two continuous processes including work and heat exchange should be considered as a combined process for analysis. For example, the isobaric condensation process 23 can be combined with the isenthalpic throttling process 34 as a combined process 24. On the other hand, the effective thermodynamic temperatures cannot be obtained by using eq. (5) for isenthalpic throttling process (Δh_{ab}=0) and isentropic compression process (Δs_{ab}=0), except with the aid of the combined process.

The effective temperature of the combined process 24 $T_{R,24}$ is also the effective thermodynamic temperature of hot reservoir $T_{R,h}$:

$$T_{R,24} = \frac{h_2 - h_4}{s_2 - s_4} = \frac{s_2 - s_3}{s_2 - s_4} T_{R,23} = T_{R,h}. \tag{8}$$

Because $s_4 > s_3$, $T_{R,24} > T_{R,h}$. Similarly, $T_{R,24}$ also can be obtained from the combined process of isentropic compression 12 and isobaric evaporation 41.

According to eqs. (1)–(4), and with the relationships of $h_2-h_4=q_h$, $h_1-h_4=q_e$, $s_1=s_2$, we can obtain the following relation by eqs. (7) and (8):

$$\frac{q_h}{q_e} = \frac{T_{R,24}}{T_{R,14}} = \frac{T_{R,h}}{T_{R,e}}. \tag{9}$$

3.3 $T''_{R,h}$ and q''_h of actual heat pump cycle

For an actual refrigeration/heat pump cycle, the isenthalpic throttling process has irreversible loss, and the isentropic compression and flowing process of working fluid also have irreversible loss. The isentropic efficiency $\eta_s=w/w''$ is usually used to describe the relation of the actual compression work w'' and the theoretical compression work w. The increasing of the w'' causes the increasing of the actual heat dissipation q''_h and the actual effective temperature of hot reservoir $T''_{R,h}$ at the same heat absorption.

$$q''_h = w'' + q_e = \Delta w + q_h = (1/\eta_s - 1)w + q_h. \tag{10}$$

The actual effective temperature of hot reservoir $T''_{R,h}$ is

$$T''_{R,h} = \frac{q''_h}{s_{14}} = \frac{q''_h}{q_h} T_{R,h} = \frac{(1/\eta_s - 1)w + q_h}{q_h} T_{R,h} = \left(\frac{(1/\eta_s - 1)w}{q_h} + 1\right) T_{R,h}. \tag{11}$$

The actual effective condensation temperature $T''_{R,23}$ is

$$T''_{R,23} = T_{Rw} + \frac{q''_h}{q_h}(T_{R,23} - T_{R,w}). \tag{12}$$

3.4 $T_{R,w}$ of hot water

If the inlet water temperature T_{w1} and outlet water temperature T_{w2} are known, q_w can be calculated as $m_w c_w(T_{w2}-T_{w1})$, and the entropy change of water is $\Delta s_{w,21}=m_w c_w \ln(T_{w2}/T_{w1})$. According to eq. (5), the effective thermodynamic temperature of hot water is defined as

$$T_{R,w} = \frac{q_w}{\Delta s_{w,21}} = \frac{T_{w2} - T_{w1}}{\ln(T_{w2}/T_{w1})}. \tag{13}$$

3.5 $T_{R,e}$ of cold air

Assuming the inlet and outlet temperatures of air are T_0 and $T_{a,1}$, the heat dissipation of air q_a (heat absorption of the evaporator q_e) is $m_a c_a(T_0-T_{a,1})$. According to eq. (5), the effective thermodynamic temperature of air cooling process is

$$T_{R,a} = \frac{q_a}{\Delta s_{a,10}} = \frac{T_0 - T_{a,1}}{\ln(T_0/T_{a,1})}, \tag{14}$$

where $\Delta s_{a,01}=m_a c_a \ln(T_0/T_{a,1})$. $T_0-T_{a,1}=q_e/m_a c_a$, which is associated with the air flux m_a and the heat absorption of the evaporator q_e. According to the optimization under the reference working conditions, the value of $(T_0-T_{a,1})_0$ is recommended as 7.5°C. If the fan speed is fixed, the relationship of $T_{a,1}$ and T_0 is

$$T_0 - T_{a,1} = 7.5\, q_e / q_{e,0}, \tag{15}$$

where $q_{e,0}$ is the heat absorption under the reference working conditions.

4 Effective thermodynamic transformation analysis method and T_R-s, T_R-h (q) diagrams

According to the parameters q_h, q''_h, $T_{R,h}$, $T''_{R,h}$, $T_{R,e}$ and T_0, we can obtain the T_R-s diagram of the equivalent reverse Carnot as shown in Figure 3, in which $\Delta s = s_2 - s_4 = q_h/T_{R,h} = q''_h/T''_{R,h} = q_e/T_{R,e}$. Using Figure 3 to replace Figure 1, it is more easy to obtain the COP of theoretical cycle and COP'' of actual cycle:

$$COP = 1/(1 - T_{R,e}/T_{R,h}), \tag{16a}$$

$$COP'' = 1/(1 - T_{R,e}/T''_{R,h}). \tag{16b}$$

Figure 4 is a combined T_R-h (q) diagram for heat pump performance analysis. The theoretical heat dissipation q_h, the actual heat dissipation q''_h, heat absorption of the evaporator q_e, the theoretical compression work w, the actual compression work w'', and all the effective thermodynamic temperatures $T_{R,h}$, $T''_{R,h}$, $T_{R,23}$, $T''_{R,23}$, $T_{R,e}$, $T_{R,w}$, $T''_{R,w}$ and $T_{R,a}$ are all shown in Figure 4. Figure 4 can be

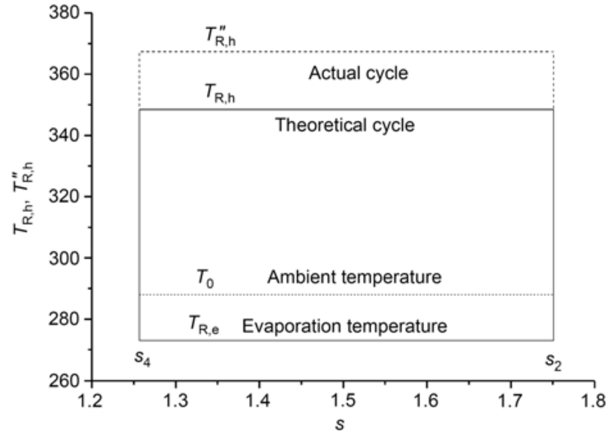

Figure 3 T_R–s diagram of the equivalent reverse Carnot.

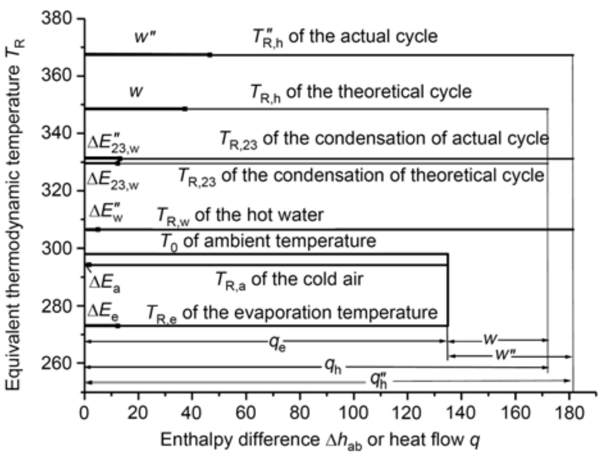

Figure 4 T_R–h (q) diagram for heat pump performance analysis.

adopted to calculate each of exergy consumptions in the exergy balance equations of theoretical cycle and actual cycle:

$$w = \Delta E_a + \Delta E_e + \Delta E_J + \Delta E_w + \Delta E_{23,w}, \tag{17}$$

$$w'' = \Delta E_a + \Delta E_{e,a} + \Delta E_J + \Delta E''_w + \Delta E''_{23,w} + \Delta E''_p, \tag{18}$$

where ΔE_a, ΔE_e, ΔE_J, ΔE_w, $\Delta E_{23,w}$ are the exergy consumptions of air flow of evaporator, heat exchange in evaporator, isenthalpic throttling process, hot water flow and heat exchange in condenser for the theoretical cycle, respectively. $\Delta E''_w$, $\Delta E''_{23,w}$ and $\Delta E''_p$ are the exergy consumptions of hot water flow, heat exchange in condenser and compression process for the actual cycle, respectively. Their calculation equations are

$$\Delta E_a = q_e\left(\frac{T_0}{T_{R,a}} - 1\right) = q_e T_0\left(\frac{1}{T_{R,a}} - \frac{1}{T_0}\right) = q_h T_0\left(\frac{1}{T_{R,a}} - \frac{1}{T_0}\right)\frac{T_{R,e}}{T_{R,h}}, \tag{19}$$

$$\Delta E_{e,a} = q_e T_0\left(\frac{1}{T_{R,e}} - \frac{1}{T_{R,a}}\right) = q_h T_0\left(\frac{1}{T_{R,e}} - \frac{1}{T_{R,a}}\right)\frac{T_{R,e}}{T_{R,h}}, \tag{20}$$

$$\Delta E_J = q_h T_0\left(\frac{1}{T_{R,23}} - \frac{1}{T_{R,h}}\right) = q_h\left(1 - \frac{T_{R,23}}{T_{R,h}}\right)\frac{T_0}{T_{R,23}}, \tag{21}$$

$$\Delta E_w = q_h\left(1 - \frac{T_0}{T_{R,w}}\right) = q_h T_0\left(\frac{1}{T_0} - \frac{1}{T_{R,w}}\right), \tag{22a}$$

$$\Delta E''_w = q''_h\left(1 - \frac{T_0}{T_{R,w}}\right) = q''_h T_0\left(\frac{1}{T_0} - \frac{1}{T_{R,w}}\right), \tag{22b}$$

$$\Delta E_{23,w} = q_h T_0\left(\frac{1}{T_{R,w}} - \frac{1}{T_{R,23}}\right), \tag{23a}$$

$$\Delta E''_{23,w} = q''_h T_0\left(\frac{1}{T_{R,w}} - \frac{1}{T''_{R,23}}\right), \tag{23b}$$

$$\Delta E''_p = \Delta E''_{ir,c} - \Delta E_J = q''_h T_0\left(\frac{1}{T''_{R,23}} - \frac{1}{T''_{R,h}}\right) - \Delta E_J, \tag{24}$$

where $\Delta E''_{ir,c}$ is the total of the exergy consumptions of the internal irreversible cycle. The values of these exergy consumptions except ΔE_J and $\Delta E''_p$, are all shown on the left of Figure 4 with bold line. It is easy to calculate $COP = q_h/w$ for the theoretical cycle, and $COP'' = q''_h/w''$ for the actual cycle.

The actual power input w' should contain the compressor power input w'', evaporator fan power consumption w_1 and condenser water pump power consumption w_2, thus,

$$w' = \Delta E_a + \Delta E_{e,a} + \Delta E''_w + \Delta E''_{23,w} + \Delta E_J + \Delta E''_p + w_1 + w_2. \tag{25}$$

Here we defined process exergy consumption ratio ξ_i as the ratio of exergy consumption of processes ΔE_i over the actual power input w'. Thus, eq. (25) can be rearranged as

$$\xi_a + \xi_{e,a} + \xi''_w + \xi''_{23,w} + \xi_J + \xi''_p + \xi_1 + \xi_2 = 1. \tag{26}$$

5 Fitting formulas of q_h, $T_{R,h}$ and $T_{R,23}$

q_h, $T_{R,h}$ and $T_{R,23}$ are three key parameters for the effective thermodynamic transformation analysis method. Although they can be obtained from the handbook or software of fluids thermodynamic properties, it is much more convenient to obtain from the fitting formulas based on the parameters of the reference cycle and the corresponding principles. For example, the fitting formulas of these parameters for R22 are

$$q_{\mathrm{h}} = 172.18\left(\frac{328}{T_3}\right)^{1.35}\left(\frac{273}{T_1}\right)^{0.8}, \tag{27}$$

$$T_{\mathrm{R},23} = \left(328+1.53\right)\left(\frac{T_3}{328}\right)\left(\frac{273}{T_1}\right)^{0.06}, \tag{28}$$

$$T_{\mathrm{R,h}} = \left[T_3 + 0.372\left(T_3-273\right)\right]\left(\frac{T_3}{328}\right)^{0.3}\left(\frac{273}{T_1}\right)^{a}, \tag{29}$$

where $a = \left[0.62 - 4.3\left(1-\frac{T_3}{328}\right) - \frac{0.08T_1}{273}\right]$.

These fitting formulas were obtained on the basis of the data of R22 in the saturated temperature range of 263 K to 328 K, which are from NIST REFPROP. The averagely relative deviations of eqs. (27)–(29) are –0.25%, 0.12%, 0.16%, respectively. The deviation for *COP* calculation is –0.010. Eqs. (27)–(29) are suitable for the evaporating temperature range of 263 K–308 K, water inlet temperature range of 278 K–308 K, and water outlet temperature range of 313 K–328 K.

The system performance can be simulated and estimated by using the fitting formulas.

6 Calculations and discussions

Through the calculation of a given example, we can demonstrate the equivalence of the proposed effective thermodynamic transformation analysis method and the traditional method, the calculation accuracy of the fitting formulas, and also can analyze the performance under variable working conditions.

The system is a heat pump water heater as Figure 1 shows, with R22 refrigerant, the inlet water temperature $T_{\mathrm{w}1}$ =286 K and outlet water temperature $T_{\mathrm{w}2}$ =328 K, isentropic efficiency η_{s}=0.8, and wind fan power w_1=0.05w. The cycle parameters are listed in Table 1, in which the thermodynamic properties are obtained from NIST REFPROP.

q_{h}, w and *COP* can be calculated by using eqs. (1)–(4) with the data in Table 1 for the theoretical cycle, and the results are 172.18 kW, 37.29 kW and 4.617, respectively.

The calculated results by using the effective thermodynamic transformation analysis method are listed in Table 2. Nos. 1 to 6 are the results of the theoretical cycle.

Nos. 1 and 3 in Table 2 are the results by using the effective thermodynamic temperature definition equations, in which w=q_{h}(1−$T_{\mathrm{R,e}}$/$T_{\mathrm{R,h}}$)=37.29 kW, *COP*=1/(1−$T_{\mathrm{R,e}}$/$T_{\mathrm{R,h}}$)= 4.617, which prove that $T_{\mathrm{R,h}}$ defined in this work is the real effective thermodynamic temperature of hot reservoir of equivalent reverse Carnot cycle. It is higher than the actual condensing temperature T_3. Moreover, there is $\sum\Delta E_i = w$ for every line in Table 2, which further proves the validity of the effective thermodynamic transformation analysis method.

Nos. 2, 4 and 6 in Table 2 are the results by using the fitting eqs. (27)–(29), the average deviation of *COP* is 0.6% with the maximum deviation of 1.4% compared with the results in Nos. 1, 3, and 5.

Nos. 5 and 6 in Table 2 are the results with the evaporating temperature T_{e} of 288 K, the values of *COP* are about 43% higher than that with the evaporating temperature T_{e} of 273 K and the same T_0. Therefore, the evaporating temperature should be optimized according to the environment temperature.

Nos. 7 and 8 in Table 2 are the results with isentropic efficiency η_{s}=0.8 and wind fan power w_1=0.05w. The calculated COP′ are similar to the actual results.

The exergy consumption ratios of Nos. 7 and 8 in Table 2 are shown in Figure 5, which gives a clear result of each process exergy consumption under different working conditions.

7 Conclusions

1) In this study, an effective thermodynamic transfor-

Table 1 Cycle parameters of R22

	1	2	2′	3	4	2″
T (K)	273	351.58	328	328	279.15	361.48
h (kJ kg^{-1})	404.99	442.28	417.64	270.1	270.1	451.60
s (kJ kg^{-1} k^{-1})	1.7509	1.7509	1.6783	1.2284	1.2568	1.7770
p (MPa)	0.49556	2.1678	2.1678	2.1678	0.49556	2.1678

Table 2 Performances calculated by effective thermodynamic temperature analysis method

No.	$T_{\mathrm{R,e}}$	$T_{\mathrm{R,a}}$	$T_{\mathrm{R,23}}$	$T_{\mathrm{R,w}}$	$T_{\mathrm{R,h}}$	T_0	$\Delta E_{\mathrm{e,a}}$	ΔE_{a}	ΔE_{w}	$\Delta E_{23,\mathrm{w}}$	ΔE_{J}	w	q_{h}	T_1	*COP*
1	273	294.23	329.53	306.52	348.47	298	10.624	1.729	4.786	11.689	8.463	37.29	172.18	273	4.617
2	273	294.23	329.53	306.52	348.46	298	10.624	1.729	4.592	11.883	8.462	37.29	172.18	273	4.617
3	273	278.31	329.53	306.52	348.47	281	2.2241	1.729	14.34	11.022	7.980	37.290	172.18	273	4.617
4	273	278.31	329.53	306.52	348.46	281	2.2241	1.729	14.34	11.021	7.976	37.286	172.18	273	4.618
5	288	294.23	328.78	306.52	339.47	298	3.067	1.794	4.588	10.866	4.708	25.02	165.05	288	6.594
6	288	294.23	328.47	306.52	338.62	298	3.075	1.799	4.588	10.724	4.484	24.67	164.97	288	6.690
No.	$T_{\mathrm{R,e}}$	$T_{\mathrm{R,a}}$	$T''_{\mathrm{R,23}}$	$T''_{\mathrm{R,w}}$	$T''_{\mathrm{R,h}}$	T_0	$\Delta E_{\mathrm{e,a}}$	ΔE_{a}	$\Delta E''_{\mathrm{w}}$	$\Delta E''_{23,\mathrm{w}}$	$\Delta E''_{\mathrm{ir,c}}$	w''	w'	q''_{h}	*COP′*
7	273	294.23	330.78	306.52	367.34	298	10.624	1.729	5.045	12.94	16.275	46.613	48.48	181.5	3.744
8	288	294.23	329.63	306.52	352.34	298	3.067	1.794	4.762	11.676	9.982	31.282	32.52	171.13	5.262

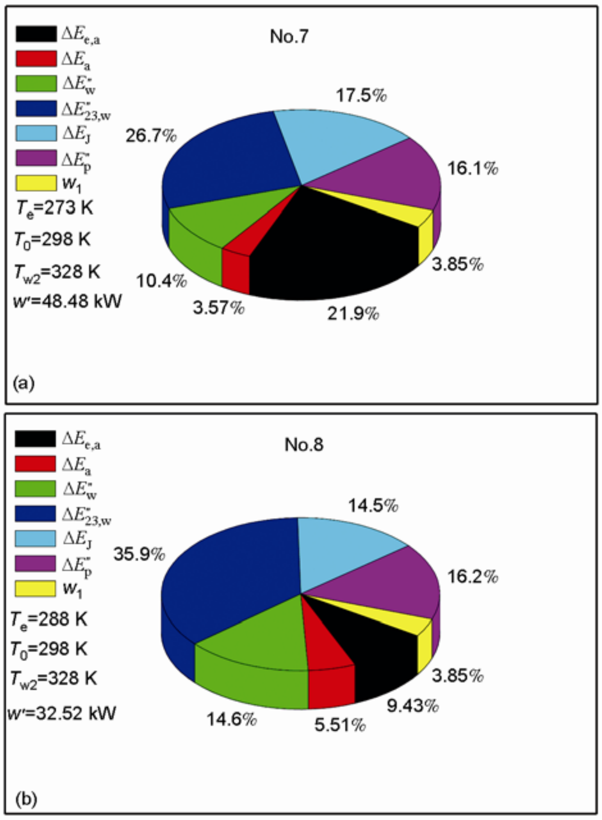

Figure 5 Exergy consumption ratios diagrams. (a) Results of No. 7 in Table 2; (b) results of No. 8 in Table 2.

mation analysis method was proposed. By means of the definitions of the effective thermodynamic temperatures of thermodynamic processes, the actual power cycle or refrigeration/heat pump cycle was transformed into the equivalent reversible Carnot or reverse Carnot cycles for thermodynamic analysis. The equations of the effective thermodynamic temperature of hot reservoir $T_{\mathrm{R,h}}$ and the actual effective temperature of hot reservoir $T''_{\mathrm{R,h}}$ of equivalent reverse Carnot cycle were derived.

2) The combined diagram of T_{R}–h and T_{R}–q was adopted for the analysis of the system performance and the exergy consumption, which takes advantage of the visual expression of the heat/work exchange and the enthalpy change, and is convenient for the calculation of the coefficient of performance and exergy consumptions.

3) By taking a heat pump water heater with refrigerant of R22 for example, the proposed method was systematically introduced, and the fitting formulas of the effective thermodynamic temperatures were given as demonstration. The results show that the proposed method has advantage and well application foreground in the performance simulation and estimation under the variable working conditions.

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