PERFORMANCE AND EXERGY ANALYSIS OF TRANSCRITICAL ORGANIC RANKINE CYCLE ASSOCIATED WITH MIXTURE WORKING FLUIDS

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ABSTRACT

The low-enthalpy heat in geothermal fluids is extracted by using a binary cycle and is converted into electricity. Conventional Organic Rankine Cycles (ORCs) are used widely in low-enthalpy geothermal power plants as an effective solution to convert low-grade heat into power. However, conventional ORCs are characterized by high exergy destruction during heat transfer in the evaporator and condenser. The irreversibility of evaporator and condenser, and exergy loss of the heat source can be reduced by employing zeotropic mixtures and transcritical organic Rankine cycles (TRCs). In the present study, a thermodynamic analysis of TRC was conducted to investigate feasibility of the organic fluids R245fa-based mixtures instead of R245fa/R134a in the low-enthalpy heat. Therefore, the mixtures, namely R245fa/R134a, R245a/R1234yf and R245fa/R290, were investigated by the first law efficiency and specific power at various inlet expander temperatures and pressures. The results indicate that the maximal specific power was occurred at temperature difference between $T_{exp,in}$ and T_{cri} ranging from 40 to 80 °C under effects of condensing temperature glide. A peak first law efficiency and specific power corresponding to a certain optimum pressure at optimal mole fraction are observed at each $T_{exp,in}$ due to effect of the irreversibilities of TRC's components. Additionally, the first law efficiency and specific power of R245fa/R290 were closed to that of R245fa/R134a. However, the operating pressure of R245fa/R290 was higher than that of R245fa/R134a. It is noted that the operating pressure was high to casue negative effects on the system. Consequently, the economic anaylysis of the mixture R245fa/R290 and R245fa/R134a will be further investigated in the future.

1. INTRODUCTION

By the growing industrialization, the accelerated consumption of fossil fuels has caused serious environmental problems and energy shortages. ORC is one of the good solutions for recovering low grade waste heat (Wei *et al.*, 2007). Tchanche *et al.* (2011) listed the ORC power plants operating, which capacity ranges from 125 kW to 6 MW. Many authors have studied ORC. The focus of the survey was on the selection of organic fluids (Chen *et al.*, 2010), (Lai *et al.*, 2011), (Saleh *et al.*, 2007) and parameter optimization (Ayachi *et al.*, 2014), (Dai *et al.*, 2009), (Schuster *et al.*, 2010).

Among several types of renewable energy sources, geothermal energy seems to be the most attractive energy source because it is stable, unaffected by weather conditions, and easily utilized by traditional technologies. However, most geothermal resources are obtained at temperatures below 150 °C and are therefore referred to as "low temperature" or "low enthalpy" geothermal resources (Barbier, 2002). In addition, the use of low temperature geothermal energy can be achieved by using a binary device. Approximately 44 % of existing units are binary plants in the geothermal due to their lower average capacity, which production is less than 10 % of the world's geothermal (Bertani, 2012).

The transcritical organic Rankine cycles (TRCs) have been identified through thermodynamics analysis as a way to improve cycle efficiency and recover more heat from waste sources. Many

research investigated thermal efficiency and exergy efficiency under similar operating conditions between subcritical and transcritical cycles. They found that the TRC improved thermal efficiency with respect to the ORC above 8 % (Vetter *et al.*, 2013), (Shu *et al.*, 2013), (Baik *et al.*, 2013).
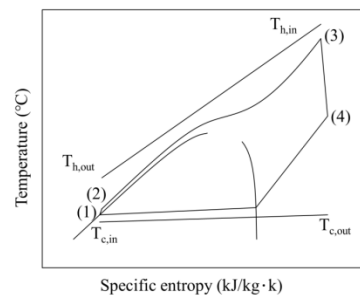
Vetter *et al.* (2013) compared to subcritical processes with isopentane, an approximately 30 % increase of specific net power output was achieved with propane as working fluid. It achieved a specific net output power of 36.8 kJ/kg and a thermal efficiency of 10.1 % at supercritical vapor. Shu *et al.* (2013) compared to the ORC system with R1234yf, the best of net output power with R143a used TRC system. The corresponding maximum values are 39.4 kW and 39.91 kW, respectively. Baik *et al.* (2012) investigated the performance of TRC using R125 and ORC using R134a, R245fa and R152a as working fluid. They found that the output power of TRC was greater than that of ORC in geothermal heat source 100 ℃. Chys *et al.* (2012) indicate that an increase in cycle efficiency, for binary mixtures, of 15.7 % and an increase in generated electricity of 12.3 % is found possible for heat source and ORC cycle parameters for a lower temperature source (150 ℃). Andreasen *et al.*, (2011) indicate that mixed working fluids can increase the net power output of the cycle, while reducing the pressure levels. In addition, the maximum net power output is obtained by fluids with a critical temperature close to half of the hot fluid inlet temperature. Gu and Sato (2002) studied transcritical cycles with propane, R125, and R134a as working fluids by using geothermal sources (>190 ℃) as heat sources, and propane and R134a were recommended for use in this cycle due to their high efficiencies. Dai *et al.* (2014) studied zeotropic mixtures of carbon dioxide blends with 7 working fluids with low GWP for use in a TRC for low-grade heat conversion. The high operation pressure can be significantly reduced, and the thermal efficiency as well as the exergy efficiency is also improved by using the mixtures as working fluids in comparison with pure CO_2. Abadi *et al.* (2015) studied the pressure ratio of the ORC with the mixture of R245fa 0.6/R134a 0.4 is lower than that of the ORC with the pure R245fa, it can generate high power using low temperature heat sources. Although hydrofluorocarbons refrigerants, namely R245a and R134a, are popular and widely used as working fluid in ORC system, they have high GWP. Hydrocarbon refrigerants (R290 and R600a), which with low GWP and higher flammability, are environmentally friendly, non-toxic and non-ozone-depleting. The objective of the present study was investigated to feasibility of the other organic fluids, namely R1234yf and R290, instead of R134a in the TRC with R245fa-based mixtures at heat source inlet temperature ranged from 160 to 210 ℃ and inlet pressure of the expander for R245fa/R134a and R245fa/R1234yf, and R245fa/R290 ranged from 4 to 8 MPa, and 5 to 9 MPa, respectively.

2. THEORETICAL ANALYSIS

The schematic diagram of TRC system is shown in Figure 1. The mixtures R245fa/R134a, R245fa/R1234yf and R245fa/R290 were used as working fluids for geothermal power plant to investigate effects of mole fraction on system performance. Figure 2 show the T-s diagram for TRC system. The figure shows the temperature glide of the working fluid vapor in the condenser. The temperature glide provids better temperature match between cooling water and working fluid during condensing process. It is meaning that the irreversibility was lower during working fluid condensing process to improve system efficiency. In the present study, the properties of the working fluid were referred to the REFPROP 9.0 and analysis system was developed by using MATLAB. The boundary conditions of TRCs associated with the mixtures and heat source are listed in Table 1.
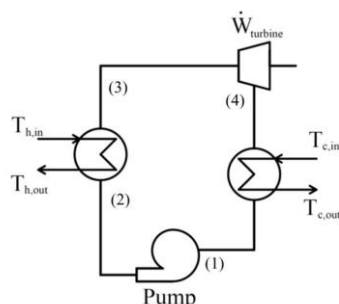
Figure 1: Schematic of the TRC system.   **Figure 2**: T-s diagram of the mixtures TRC systems.

For the TRC system the pumps pressurize the working fluid to a supercritical state, resulting in a high pressure difference between the evaporator and condenser. However, the working fluid pumps consume considerable power from the system generator. To understand the weighting of the consumed power on the output power of the system, the net thermal efficiency (first law efficiency) is estimated using the following equations:

$$\dot{Q}_{hs} = \dot{m}_f(h_3 - h_2) \tag{1}$$

$$\dot{W}_{pump} = \dot{m}_f(h_2 - h_1) \tag{2}$$

$$\dot{W}_{exp} = \dot{m}_f(h_3 - h_4) \tag{3}$$

$$\dot{P}_{gen} = \dot{W}_{exp} \cdot \eta_{g,m} \tag{4}$$

$$\dot{W}_{net} = \dot{W}_{exp} - \dot{W}_{pump} \tag{5}$$

$$\zeta = \frac{\dot{P}_{gen}}{\dot{m}_{hs}} \tag{6}$$

$$\eta_I = \frac{\dot{W}_{net}}{\dot{Q}_{hs}} \tag{7}$$

$$\dot{E}_{hs} = \dot{m}_{hs}[(h_{hs,in} - h_{hs,out}) - T_0(S_{hs,in} - S_{hs,out})] \tag{8}$$

$$\dot{E}_{cs,out} = \dot{m}_{cs}[(h_{cs,out} - h_{cs,in}) - T_0(S_{cs,out} - S_{cs,in})] \tag{9}$$

$$\dot{E}_{hS} = \dot{E}_{cs,out} + \dot{I}_{total} + \dot{W}_{net} \tag{10}$$

Where $\dot{W}_{net}$ is the difference between the shaft power produced by the expander and the power consumed by the pump of working fluid. The total heat transfer rate in the evaporator is a product of the mass flow rate of the working fluid and enthalpy difference between the evaporator outlet and inlet.

The efficiency of the main components and the system was evaluated using thermodynamic and energy balance equations. The main design points and components are displayed in Figure 2. The mathematical model for each component is described as follows.

The exergy destruction of the pumping process between point 1 and point 2 is defined as:

$$\dot{I}_{pump} = \dot{m}_f T_o(S_2 - S_1) \tag{11}$$

$$\Omega_{pump} = \frac{\dot{I}_{pump}}{\dot{E}_{hs}} \tag{12}$$

The exergy destruction of the evaporating process between point 2 and point 3 is defined as:

$$\dot{I}_{eva} = \dot{m}_f T_o(S_3 - S_2) - \dot{m}_{hs} T_o(S_{hs,in} - S_{hs,out}) \tag{13}$$

$$\Omega_{eva} = \frac{\dot{I}_{eva}}{\dot{E}_{hs}} \tag{14}$$

The exergy destruction of the expanding process between point 3 and point 4 is defined as:

$$\dot{I}_{exp} = \dot{m}_f T_o(S_4 - S_3) \tag{15}$$

$$\Omega_{exp} = \frac{\dot{I}_{exp}}{\dot{E}_{hs}} \tag{16}$$

The exergy destruction of the condensing process between point 4 and point 1 is defined as:

$$\dot{I}_{cond} = \dot{m}_f T_0 (S_4 - S_1) - \dot{m}_{cs} T_0 (S_{cs,in} - S_{cs,out}) \tag{17}$$

$$\Omega_{cond} = \frac{\dot{I}_{cond}}{\dot{E}_{hs}} \tag{18}$$

The sum of the exergy destruction is defined as:

$$\dot{I}_{total} = \dot{I}_{pump} + \dot{I}_{eva} + \dot{I}_{exp} + \dot{I}_{cond} \tag{19}$$

$$\Omega_{total} = \Omega_{pump} + \Omega_{eva} + \Omega_{exp} + \Omega_{cond} \tag{20}$$

Table 1: Given parameters of TRCs

Turbine isentropic efficiency	η_{exp}	80 %
Pump isentropic efficiency	η_{pump}	65 %
Generator/motor efficiency	$\eta_{g,m}$	95 %
Temperature difference of pinch point in the evaporator	$\Delta T_{pp,eva}$	5 K
Temperature difference of pinch point in the condenser	$\Delta T_{pp,cond}$	5 K
Inlet temperature of the expander	$T_{exp,in}$	150-200 °C
Temperature of Geothermal fluid	$T_{hs,in}$	160-210 °C
Mass flow rate of Geothermal fluid	$\dot{m}_{hs}$	30 kg/s
Inlet temperature of Cooling source	$T_{cs,in}$	32 °C
Ambient temperature	T_0	32 °C

3. RESULTS AND DISCUSSION

In the present study, R245fa was mixed with R134a, R1234yf and R290, respectively. Although R134a, R1234yf and R290 have the similar critical point, GWP of R134a is higher than that of R1234yf and R290. The thermodynamic and environmental properties of the pure fluid are listed in Table 2. The present study was carried out theoretical analysis to investigate feasibility of the organic fluids R245fa-based mixtures instead of R245fa/R134a in the TRC with mixtures. For R245fa/R134a, R245fa/R1234yf and R245fa/R290 the optimal mole fraction and optimal expander inlet pressure corresponding to the maximal first law efficiency of thermodynamics and maximal specific power at various inlet temperatures are listed in Table 3. For R245fa/R134a and R245fa/R1234yf the variation of mole fraction with $T_{exp,in}$ had similar trend. However, for R245fa/R290 the mole fraction was fixed at 0.41/0.59 at given $P_{exp,in}$, except $T_{exp,in} = 200$ °C. As $T_{exp,in} \leq 180$ °C, the mole fraction of R245fa with high critical temperature was low. This was because of the maximal specific power occurred at temperature difference between $T_{exp,in}$ and T_{cri} ranging from 40 to 80 °C under effects of condensing temperature glide.

Table 2: Thermodynamic properties and environmental data of working fluids.

Substance	Physical data				Environmental data			Fluid type
	Molecular mass	T_b (°C)	T_{cri} (°C)	P_{cri} (MPa)	Atmospheric life (year)	ODP	GWP	

	(g/mol)							
R245fa	134.05	15.14	154.01	3.65	7.60	0	1030	Dry
R134a	102.03	-26.07	101.10	4.06	14.00	0	1370	Wet
R1234yf	114.04	-29.45	94.70	3.38	0.03	0	<4.4	Dry
R290	44.10	-42.00	96.69	4.25	0.04	0	~20	Wet

Table 3: Mole fraction of the mixtures and expander inlet pressure (in the round brackets) corresponding to the maximal first law efficiency of thermodynamics and maximal specific power.

$T_{exp,in}$		150 ℃	160 ℃	170 ℃	180 ℃	190 ℃	200 ℃
R245fa/R134a	η_I	0.16/0.84	0.16/0.84	0.16/0.84	0.17/0.83	0.95/0.05	1/0
		(5.8 MPa)	(6.5 MPa)	(7.2 MPa)	(8.0 MPa)	(8.0 MPa)	(8.0 MPa)
	ζ	0.17/0.83	0.17/0.83	0.17/0.83	0.17/0.83	0.95/0.05	095/0.05
		(7.0 MPa)	(7.4 MPa)	(7.9 MPa)	(8.0 MPa)	(8.0 MPa)	(8.0 MPa)
R245fa/R1234yf	η_I	0.2/0.8	0.2/0.8	0.2/0.8	0.2/0.8	0.95/0.05	1/0
		(5.5 MPa)	(6.2 MPa)	(7.0 MPa)	(7.8 MPa)	(8.0 MPa)	(8.0 MPa)
	ζ	0.2/0.8	0.2/0.8	0.2/0.8	0.24/0.76	0.97/0.03	0.97/0.03
		(6.4 MPa)	(6.8 MPa)	(7.3 MPa)	(7.6 MPa)	(7.6 MPa)	(8.0 MPa)
R245fa/R290	η_I	0.41/0.59	0.41/0.59	0.41/0.59	0.41/0.59	0.41/0.59	1/0
		(6.3 MPa)	(7.0 MPa)	(7.8 MPa)	(8.6 MPa)	(9.0 MPa)	(9.0 MPa)
	ζ	0.41/0.59	0.41/0.59	0.41/0.59	0.41/0.59	0.41/0.59	1/0
		(6.6 MPa)	(7.2 MPa)	(7.8 MPa)	(8.4 MPa)	(8.9 MPa)	(9.0 MPa)

The total normalized exergy destruction for the mixtures was numerically examined at various the inlet pressures of the expander, as shown in Fig. 3. At a relative low inlet temperature of the expander, the exergy destruction was decreased firstly and then increased with an increase of $P_{exp,in}$. When $T_{h,in}$ of heat source was lower than 180 ℃, the evaporating pinch temperature was higher than critical temperature of the mixtures. Consequently, the irreversibility of evaporate could not notably be improved by the operating pressure resulted in temperature mismatch between heat source and working fluid in evaporator.

(a) R245fa 0.17/R134a 0.83   (b) R245fa 0.2/R1234yf 0.8   (c) R245fa 0.41/R290 0.59
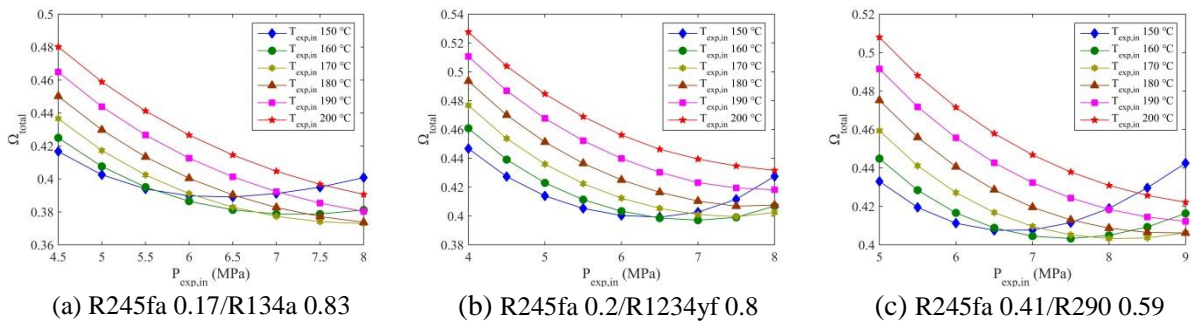
Figure 3: Effect of $T_{exp,in}$ and $P_{exp,in}$ on the normalized exergy destruction of the mixtures.

Figure 4 and 5 show that the influence of $P_{exp,in}$ and $T_{exp,in}$ on the first law efficiency and specific power of of the mixtures at a given mole fraction. The first law efficiency and specific power were considerably increased with $T_{exp,in}$. Furthermore, at relative low $P_{exp,in}$, the first law efficiency was slightly increased with $T_{exp,in}$. By contrast, specific power was considerably improved by the $T_{exp,in}$ and slightly affected by $P_{exp,in}$. Meanwhile, the peak value of the first law efficiency and specific power corresponding to a certain optimum pressure are observed at each $T_{exp,in}$. These optimal pressures for the peak effeciency and power were varied with the various $T_{exp,in}$. This was because of the irreversibility of evaporator and condenser firstly decreased with an increase of $P_{exp,in}$ and then approached to the lowest value. However, the irreversibility of the expander and pump was linearly increased with $P_{exp,in}$. For relative high $P_{exp,in}$ the irreversibility of the expander and pump was higher than that of evaporator and condenser resulted in reduced the first law efficiency and specific power.

From the analytical results the mixtures had similar behaviour in the efficiency and output power at various $T_{exp,in}$ and $P_{exp,in}$. Furthermore, the optimum pressure of R245fa/R290 was notably higher than

that of R245fa/R1234yf and R245fa/R134a (as shown in Table 3, Figs. 4 and 5). For example, at the inlet temperature of 200 °C, the optimum pressure of R245fa/R134a, R245fa/R1234yf and R245fa/R290 were 8, 8 and 9 MPa and the corresponding maxumum first law efficiency were 13.6, 12.7 and 12.9 %, respectively. The corresponding specific powers for R245fa/R134a, R245fa/R1234yf and R245fa/R290 were 86, 79 and 81 kJ/kg, respectively. It can be concluede that performance of R245fa/R290 was obviously approached to that of R245/R134a. However, operating pressure of R245fa/R290 was higher than that of R245fa/R134a. The high operating pressure of the system was negative effects on the system in terms of cost, safety and consumed power of pump. Consequently, the economic anaylysis of the mixture R245fa/R290 and R245fa/R134a will be further investigated in the future.
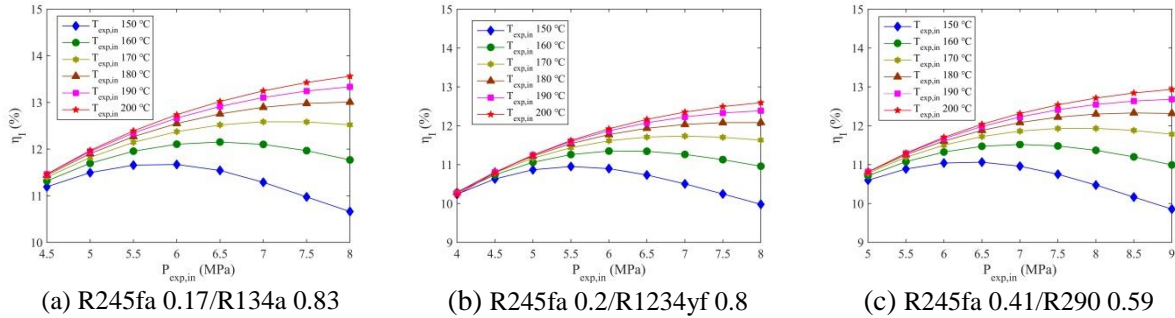
| (a) R245fa 0.17/R134a 0.83 | (b) R245fa 0.2/R1234yf 0.8 | (c) R245fa 0.41/R290 0.59 |

Figure 4: Effect of $T_{exp,in}$ and $P_{exp,in}$ on the First law efficiency of the mixtures.

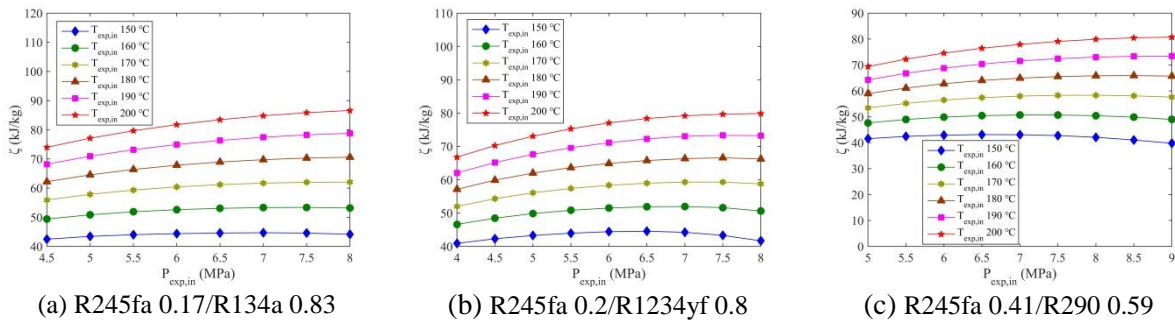| (a) R245fa 0.17/R134a 0.83 | (b) R245fa 0.2/R1234yf 0.8 | (c) R245fa 0.41/R290 0.59 |

Figure 5: Effect of $T_{exp,in}$ and $P_{exp,in}$ on the specific power of the mixtures.

6. CONCLUSIONS

In the present study, the mixtures (R245fa/R134a, R245fa/R1234yf and R245fa/R290) with optimal mole fraction was examined by thermodynamic method at various inlet expander pressures and temperatures to investigate feasibility of the other organic fluids, namely HC and HFO, instead of HFC in the TRC with mixtures. The conclusions were summarized as follows:

(1) The optimal mole fraction of the mixtures corresponding to maximal first law efficiency and specific power was notably affected by $T_{exp,in}$ and $P_{exp,in}$. The peak value of the first law efficiency and specific power was appeared at temperature difference between $T_{exp,in}$ and T_{cri} ranging from 40 to 80 °C under effect of condensing temperature glide.

(2) A peak first law efficiency and specific power corresponding to a certain optimum pressure were observed at each $T_{exp,in}$. These optimal pressures for the peak effeciency and power were varied with the various $T_{exp,in}$.

(3) For R245/R290 the first law efficiency and specific power were apporached to that of R245fa/R134a. It can be seen that R245fa/R290 will have high potential to replace R245fa/R134a in the TRC system. However, R245fa/R290 had the highest operating pressure to cause problem of cost, safety and consumed pumping power. Meanwhile, the economic anaylysis of the mixture R245fa/R290 and R245fa/R134a will be further investigated in the future.

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NOMENCLATURE

$\dot{E}$	exergy flow rate (kW)
h	specific enthalpy (kJ/kg)
$\dot{I}$	flow rate of exergy destruction (kW)
$\dot{m}$	mass flow rate (kg/s)
P	pressure (MPa)
$\dot{Q}$	heat transfer flow rate (kW)
s	specific entropy (kJ kg^{-1} K^{-1})
T	temperature (℃)
$\dot{W}$	power input/output (kW)

Greek letters

η	efficiency (%)
ζ	specific power (kJ/kg)
Ω	irreversibility rate
ΔT	temperature difference (℃)

Subscript

cs	cooling source
cs,in/cs,out	cooling source inlet/outlet
cri	critical
cond	condenser/condensation
c,in/c,out	cooling source inlet/outlet
eva	evaporator/evaporation
exp	expander
gen	generator
hs	heat source
hs,in/hs,out	heat source inlet/outlet
net	net
pp	pinch point
f	working fluid
I	first law efficiency of thermodynamic
0	environment state
in/out	inlet/outlet

Acronyms

ODP	ozone depletion potential
GWP	global warming potential
HC	hydro carbon
HFO	hydrofluoroolefin
HFC	hydro fluorine carbon