

Performance analysis of a new deep super-cooling two-stage organic Rankine cycle

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Abstract

In this article, a new deep super-cooling two-stage organic Rankine cycle (DTORC) is proposed and evaluated at high temperature waste heat recovery in order to increase the power output. A thermodynamic model of recuperative organic rankine cycle (ORC) is also established for the purpose of comparison. Furthermore, a new evaluation index, effective heat source utilization, is proposed to reflect the relationship among the heat source, power output and consumption of the waste heat carrier. A simulation model is formulated and analysed under a wide range of operating conditions with the heat resource temperature fixed at 300 °C. Hexamethyldisiloxane (MM) and R245fa are used as the working fluid for DTORC, and MM for ORC. In the current work, the comparisons of heat source utilization, net thermal efficiency as well as the total surface area of the heat exchangers between DTORC and RC are discussed in detail. Results show that the DTORC performs better than ORC at high temperature waste heat recovery and it could increase the power output by 150%. Moreover, the maximum net thermal efficiency of DTORC can reach to 23.5% and increased by 30.5% compared with that using ORC, whereas the total surface areas of the heat exchangers are nearly the same.

Keywords: High temperature waste heat; deep super-cooling; two-stage; organic Rankine cycle; effective heat source utilization.

Nomenclature

h	specific enthalpy, kJ/kg	p	pump
\dot{m}	mass flow rate, kg/s	$pinch$	pinch point temperature difference
P	pressure, kPa	rec	recuperate
Q	quantity of heat, kJ	s	isentropic state
t/T	temperature, °C	sys	the whole system
W	power, kW	set	user settings
Sur_{total}	the total surface area of heat exchangers, m ²	sur	surrounding
η_{net}	net thermal efficiency, %	$subcool$	subcooled status
$\eta_{utilization}$	heat source utilization, %	MM	the MM system
		$R245fa$	the R245fa system
		$1-23$	points on plant scheme or in $T-s$ diagram
Greek letters			
η	efficiency, %		
Subscripts		<i>Abbreviations</i>	
ab	absorb	BORC	basic organic Rankine cycle
c	critical state	CHP	combined heat and power
dis	discharge	DTORC	deep super-cooling two-stage organic Rankine cycle
exp	expander	EHE	external heat exchanger
HTF	heat transfer fluid	HTF	heat transfer fluid
$HTFin$	the heat transfer fluid inlet the evaporator	ICE	internal combustion engine
$HTFout$	the heat transfer fluid outlet the evaporator	IHE	inner heat exchanger
$loss$	loss	MM	hexamethyldisiloxane
$max1$	the maximum of first stage	MDM	octamethyltrisiloxane
$max2$	the maximum of second stage	ORC	organic Rankine cycle
$min1$	the minimum of first stage	RC	recuperative ORC
net	net	R245fa	perfluoropropane
		TORC	two-stage ORC
		WHC	waste heat carrier

29

30 1. Introduction

31 With rapidly increasing globalization and energy demands, researchers are devoted to
 32 seeking opportunities to improve the energy efficiency. Among the energy-efficient technologies,
 33 organic Rankine cycle (ORC) is considered as one of the most attractive ways to solve the
 34 energy crisis in the future [1]. Over the past two decades, a number of research efforts have been

35 devoted to the studies of ORC on solar thermal [2, 3], geothermal [4, 5], internal combustion
36 engine(ICE) [6, 7], combustion gas turbine [8, 9], combined heat and power (CHP) [10], waste
37 heat from power plants [11] and industrial processes [12, 13]. For those above application, it is
38 recognized that the heat source temperature is usually in the range of 250-350 °C. Gao et al. [3]
39 developed an ORC system driven by solar energy and the high temperature of the system can be
40 up to 300 °C. Sarkar [14] pointed out that the waste heat at temperatures can be 300-400 °C in
41 some industries such as iron and steel, glass, nonferrous metals, bricks and ceramics processing.
42 Peris et al. [12] summarized the wide range of the waste heat source temperatures from industrial
43 gases and over 60% were in the temperature range of 250-350 °C.

44 For the case of high temperature waste heat recovery applications, thermal systems should
45 have a high thermal stability requirement against the working fluid. Nowadays, most of the ORC
46 manufacturers [15] and researchers [3, 16] are using siloxanes as the working fluids as they have
47 the desired characteristics in reaching high working temperatures and are more thermally stable
48 and environmentally friendly. However, siloxanes are dry organic fluids [17] and have a large
49 sensible heat in isobaric heat discharging sub-process, therefore, recuperative ORC is used in the
50 existing systems. It can significantly improve the thermal efficiency of the system, however, it is
51 difficult to improve the utilization level of the high temperature waste heat sources.

52 In the ORC system, the inner heat exchanger (IHE) leads to a high exhaust temperature of
53 the waste heat carrier (WHC) [18, 19]. The WHC can be a fluid, steam or exhaust gases and
54 often forms an open loop. The IHE only shifts part of the sensible heat in the isobaric heat
55 discharging sub-process from the working fluid to the WHC. This sensible heat will not be
56 converted into power rather will be discharged to the environment carried by the WHC. For a
57 certain waste heat source, ORC system may consume much more waste heat carrier than a basic

58 ORC (BORC) system under the same power output since the BORC can absorb more heat from
59 the unit mass flow rate of the waste heat carrier. In such situations, the extra consumption of the
60 mass flow rate of the waste heat carrier can lead to the resource wastage.

61 Compared with IHE, the external heat exchanger (EHE) can also take advantage of the heat
62 in the isobaric heat discharging sub-process. EHE can transfer such heat to an extra ORC acting
63 as an evaporator. This system is called two-stage organic Rankine cycle (TORC). Kosmadakis et
64 al. [20] introduced a TORC for reverse osmosis (RO) desalination. In their work the condenser
65 of the high temperature stage acted as the evaporator of the low temperature stage. They used
66 R245fa at high temperature stage and HFC-134a at low temperature stage. Xue et al. [21]
67 proposed a TORC using R227ea and R116 as the working fluids at high and low temperature
68 stage, respectively. Thierry et al. [22] studied the mixtures as the working fluid in TORC, and
69 improved the amount of energy recovery. All these TORC mentioned above are used at low
70 temperature waste heat recovery and have not been applied at high temperature.

71 Furthermore, how to evaluate the utilization of the heat source and the performance of the
72 thermal systems is also an important issue. Over the past decades, evaluation indexes are mainly
73 concentrated on the economic cost, thermal efficiency and exergy efficiency [23-27]. Some
74 researchers also take the maximum power output as the optimization goal [28, 29]. However,
75 few research works take the WHC consumption into account for the performance evaluation. In
76 particular, no evaluation index is available to reflect the relationship among the heat source,
77 power output and WHC consumption. As such, the objective of the current study is to develop a
78 deep super-cooling two-stage organic Rankine cycle (DTORC) based on the traditional TORC
79 with fully utilize the high temperature waste heat and increase the power output. In addition, a
80 new evaluation index, effective heat source utilization, is also defined which will evaluate the

94 As the DTORC is used for high temperature waste heat recovery, hexamethyldisiloxane
 95 (MM) has been selected for the high temperature stage due to its appropriate thermodynamic
 96 properties and optimum thermal stability. The normal boiling point of MM is 100.25 °C. This
 97 means the outlet temperature of the high temperature stage turbine can reach 203 °C or higher if
 98 the condensation pressure remains positive, as shown in Fig. 2. According to the research of [30],
 99 it shows that R245fa is more suitable as the working fluid under such a temperature compared to
 100 other working fluids, thus, the refrigerant R245fa has been selected for the low temperature stage.
 101 Table 1 lists the characteristics of the working fluids for the DTORC.

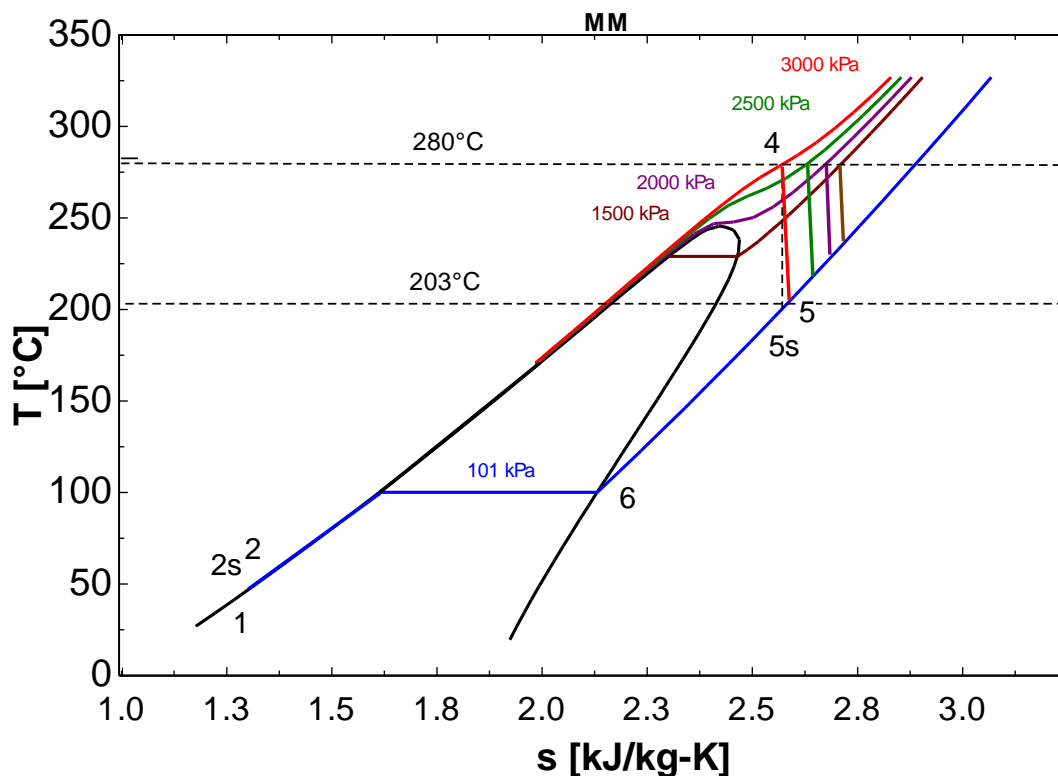


Fig. 2. T-s diagram of MM.

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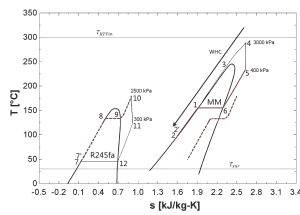
108 **Table 1**
 109 Characteristics and properties of MM and R245fa.

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fluid	M (g/mol)	T_{NBP} (°C)	T_C (°C)	P_C (kPa)	ODP	GWP (100 years)	Type	Price (\$/10kg)
MM	162,378	100.25	245.55	1939.4	0	0	Dry	61
R245fa	134,045	15.14	154.01	3651.0	0	1030	Dry	251

111 NBP: Normal boiling point; C: Critical.

112



113

114 **Fig. 3.** T - s diagram of MM and R245fa.

115 In the present work, a new deep super-cooling technology was proposed, as shown in Fig. 3.

116 The sub-process (1-2) is the deep super-cooling technology. The processes in DTORC are

117 described as follows:

118 2'-3-4: The working fluid MM absorbs the heat from the WHC at high temperature stage.
119 4-5: The supercritical steam of MM enters the turbine to produce power.
120 5-6-1-2: MM discharges the heat to the low temperature stage. In the traditional TORC
121 cycle, MM discharges heat and ends at point 1. For the current cycle, the low temperature
122 stage makes MM run till the deep super-cooling status and ends at point 2.
123 2-2': The deep super-cooling MM is pumped to a high pressure by the feed pump.
124 7'-8-9-10: The working fluid R245fa of the low temperature stage absorbs the heat from the
125 high temperature stage.
126 10-11: The overheating steam of R245fa enters the turbine to produce power.
127 11-12-7: R245fa discharges the heat to the environment.
128 7-7': The liquid R245fa is pumped to a high pressure by the feed pump.

129 Through the deep super-cooling process (1-2), MM achieves a lower temperature before
130 entering the evaporator. This makes the unit mass of WHC discharge more heat and improves
131 the utilization of the high temperature waste heat. With the aid of this low temperature stage the
132 whole system achieves an additional power output and the overall efficiency increases
133 significantly.

134 **3. System Modeling**

135 In the current work, numerical models of the DTORC (see Fig. 1) and RC (see Fig. 4)
136 systems have been developed using Engineering Equation Solver (EES) which provides an
137 inner fluid property database including those of MM and R245fa. The RC system is selected for
138 the purpose of comparison.

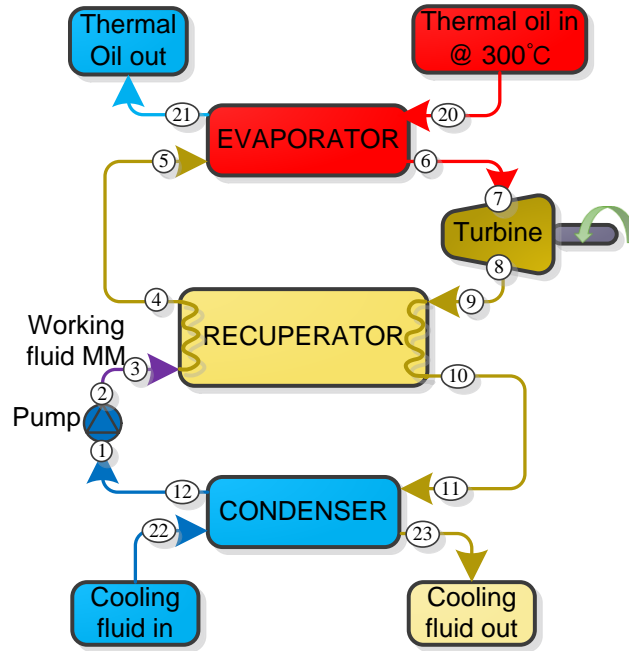


Fig. 4. Schematic diagram of RC system.

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141 *3.1. Assumptions and index defined*

142 Thermal oil (THERMINOL® 66, Solutia Inc.) is selected to imitate the WHC. The thermal
 143 oil is also called heat transfer fluid (HTF). Some thermodynamic properties of the thermal oil,
 144 such as density and enthalpy, are available at temperatures ranging from -7 °C to 371 °C as
 145 stated by Solutia Inc. In addition, some assumptions are made as follows:

146 i. The waste heat carrier has a highest inlet temperature (T_{HTFin}) of 300 °C, and the
 147 surrounding temperature (T_{sur}) is assumed to be 30 °C.

148 ii. The pressure drop is fixed at 20 kPa both in the evaporator and condenser which can be
 149 ignored in the pipe. The minimum pressure of DTORC and RC remains positive to prevent air
 150 infiltration.

151 iii. The heat transfer coefficients of the DTORC and RC systems are fixed at 3500 W/m²K
 152 by reference to an ordinary plate heat exchanger. It means the total heat transfer area is
 153 proportional to the total amount of heat transfer.

154 iv. The isentropic efficiencies of the pump and expander (η_{exp} & η_p) are considered as
155 constant of 80% and the expansion and compression processes have been assumed to be
156 adiabatic.

$$157 \quad \eta_p = (h_{2s} - h_1) / (h_2 - h_1) \quad (1)$$

158 where h_1 and h_2 are the specific enthalpies at the inlet and outlet of the pump respectively
159 (see Fig. 2), and h_{2s} is the specific enthalpy following an isentropic pumping process that starts
160 from the same state at the pump inlet and ends at the same pressure at the pump outlet.

$$161 \quad \eta_{exp} = (h_4 - h_5) / (h_4 - h_{5s}) \quad (2)$$

162 where h_{5s} is the specific enthalpy at the expander exit for an isentropic process (see Fig. 2).

163 v. The pinch temperatures (t_{pinch}) in DTORC and RC are more than 18 °C and all the
164 working fluids remain a superheat of 2 °C to flow into the expander or at least 20 kPa lower than
165 the saturation pressure under the expander inlet temperature.

166 vi. All systems are modeled under steady state conditions based on the first principle of
167 energy conservation. The entire net power outputs of both DTORC and RC systems are fixed at
168 200 kW.

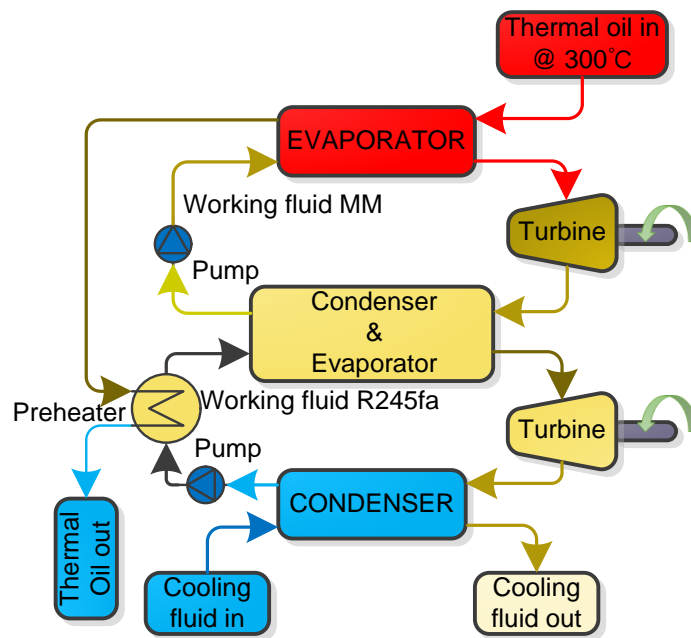
169 Besides, the heat source utilization ($\eta_{utilization}$) is defined as the ratio of the net power
170 output to the total thermal energy of the heat source in which $T_{HTF_{in}}$ and T_{sur} are considered as
171 the reference temperatures:

$$172 \quad \eta_{utilization} = W_{net} / ((h_{HTF, T_{HTF_{in}}} - h_{HTF, T_{sur}}) * \dot{m}_{HTF}) \quad (3)$$

173 here $h_{HTF, T_{HTF_{in}}}$ and $h_{HTF, T_{sur}}$ are the specific enthalpies of the waste heat carrier at the
174 temperature of $T_{HTF_{in}}$ and T_{sur} respectively (see Fig. 3), and \dot{m}_{HTF} is the mass flow rate of the
175 WHC. W_{net} is the net power output of the thermal system.

176 *3.2. DTORC system modeling*

177 It is noted that if the deep super-cooling process (1-2) of MM (see Fig. 3) is abandoned and
 178 the WHC (thermal oil) leaving the high temperature stage is used to preheat R245fa directly as
 179 shown in Fig. 5, the result of the emulational modeling would be divergent. As a result, the
 180 current work mainly focuses on the system demonstrated in both Fig. 1 and Fig. 3. For the
 181 described cycle, the simulation is performed using an EES-based program. The enthalpy of the
 182 THERMINOL® 66 is programmed to be an inner function of EES based on the data provided by
 183 manufacturer.



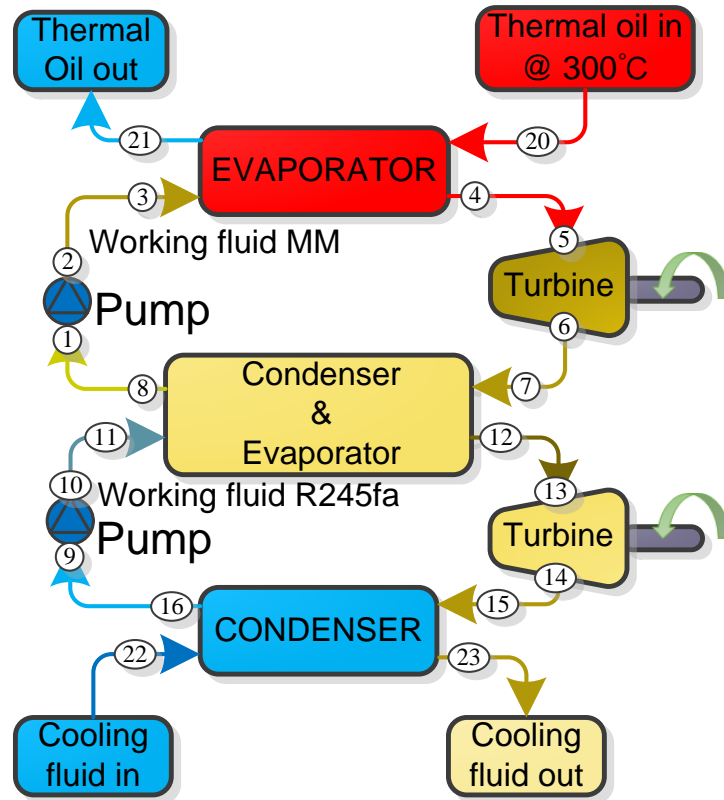
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 185 **Fig. 5.** Schematic diagram of TORC with preheating.

186 *3.2.1. Two notes in the program*

187 In this section, two key factors should be stressed here:

188 Note 1: as shown in Fig. 3, the process (11-12) in the low temperature stage can also satisfy
 189 the inner heat transfer condition which is similar to Fig. 6(b), and the proposed modelling has
 190 been designed to determine whether to use a recuperator or not at low temperature stage
 191 according to the temperatures at point 14 and point 10 (see Fig. 6a). If the temperature of point

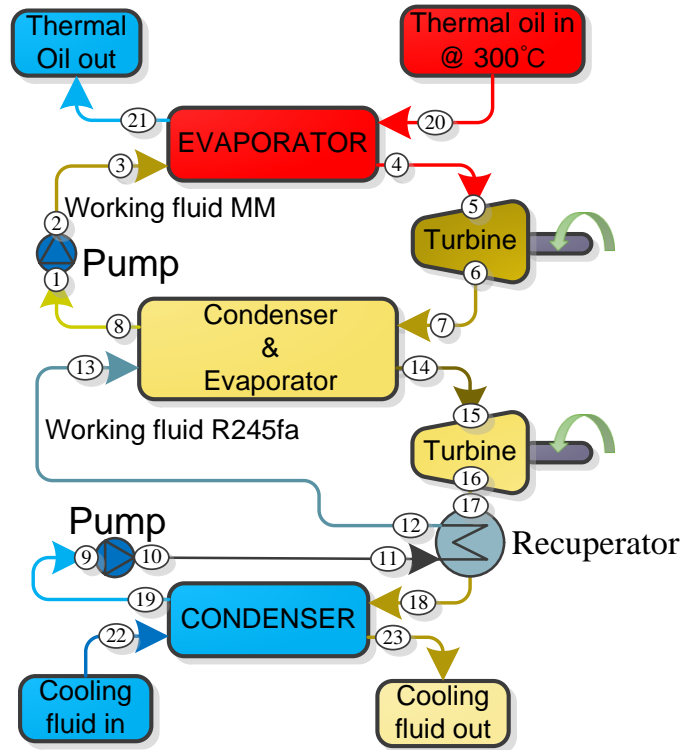
192 14 is one t_{pinch} higher than that of point 10, then the model will be changed from Fig.6 (a) to Fig.
 193 6(b). As a matter of fact, the use of a recuperator should be avoided at low temperature stage,
 194 since it will increase the pump inlet temperature at high temperature stage thus leading to a rise
 195 of the evaporator inlet temperature. This will increase the emission temperature of the waste heat
 196 carrier and may cause a decline of the heat source utilization. Therefore, isentropic or wet
 197 working fluid is suggested for the low temperature stage. Although R245fa is a dry working fluid,
 198 its saturated vapor line is very close to that of an isentropic fluid and its temperature profile is
 199 suitable for working with MM.



200

201

(a)



(b)

Fig. 6. Schematic diagram of DTORC system at low temperature stage:

(a) without recuperator; (b) with recuperator.

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207 Note 2: the temperature difference of heat exchange between the process (7-8) and (11-12)

208 (see Fig. 6(a)) should always satisfy t_{pinch} . For example, as shown in Fig. 7, the temperatures at

209 point 7 and 8 are higher than that at point 12 and 11, respectively, but the vaporization

210 temperature of the R245fa cycle is higher than the liquefaction temperature of the MM cycle.

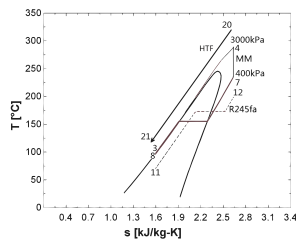
211 This may result in an impractical heat transfer in the thermodynamic model and it should be

212 avoided. In the current study, this can be solved by adding a constraint between the condensation

213 pressure of the MM cycle and the evaporation pressure of the R245fa cycle, and making sure that

214 the evaporating temperature of R245fa is at least one t_{pinch} lower than the condensing

215 temperature of MM.



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Fig. 7. An impractical heat transfer process between two stages in DTORC.

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3.2.2. Flowchart of TORC

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The simplified block diagram in Fig. 8 illustrates the basic concept of the DTORC System modeling. In the current modeling, $P_{set,max1}$, $P_{set,min1}$, $P_{set,max2}$, $P_{set,min2}$ and $t_{set,subcool,1}$ are the input parameters. Normally, P_{max1} (pressure after pump in the high temperature stage), P_{min1} (pressure before pump in the high temperature stage), P_{max2} (pressure after pump in the low temperature stage), P_{min2} (pressure after pump in the low temperature stage), and $t_{subcool,1}$ (super-cooling degree of MM in the high temperature stage) are equal to the input parameters. However, in some cases, the input parameters are in conflict with the boundary conditions, then the “**Protection program**” will limit P_{max1} , P_{min1} , P_{max2} , P_{min2} , and $t_{subcool,1}$ in a reasonable

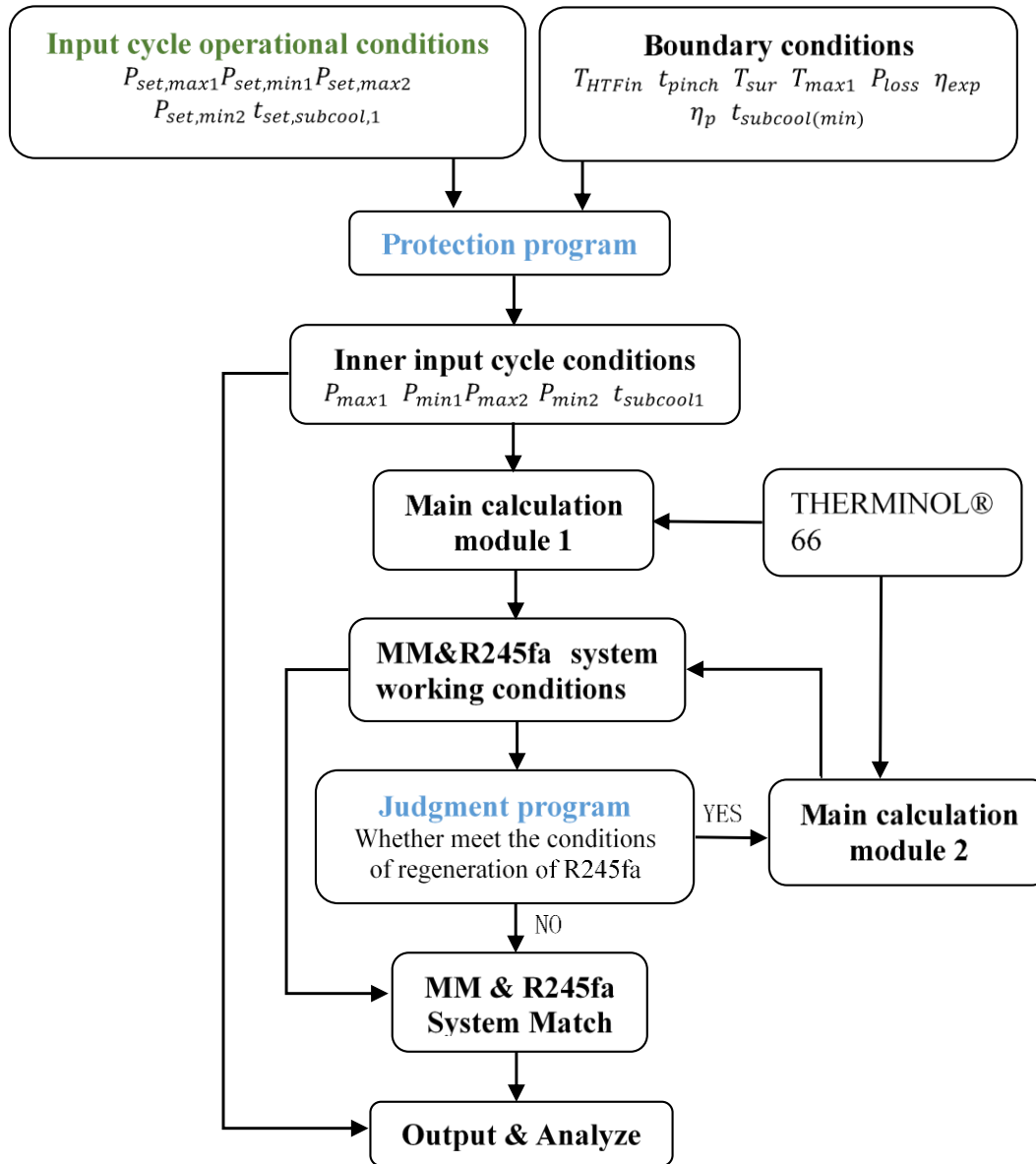
227 range. Meanwhile, P_{max1} , P_{min1} , P_{max2} , P_{min2} and $t_{subcool,1}$, which are determined by the
228 anticipant input parameters and boundary conditions, are presented in an clear manner.

229 As shown in Fig. 3:

$$230 \quad t_{subcool,1} = T_1 - T_2 \quad (4)$$

231 There is a maximum $t_{subcool,1}$ to make T_2 one t_{pinch} higher than $T_{7'}$, which is the pump
232 outlet temperature of the R245fa cycle. When the anticipant $t_{set,subcool,1}$ exceeds the maximum
233 $t_{subcool,1}$, the “**Protection program**” will use the maximum $t_{subcool,1}$; $P_{set,max1}$ is the anticipant
234 pump outlet pressure of MM ($P_{2'}$); If the “**Protection program**” detects that the state of MM at
235 point 4 does not satisfy assumption(v), the program will limit P_4 to the maximum pressure
236 under T_4 and change the value of P_{max1} ; P_{min1} (P_2) and P_{min2} (P_7) should be positive to prevent
237 air infiltration; P_{max2} ($P_{7'}$) should satisfy both assumption(v) and Note 2.

238 Note 1 will be achieved automatically by the “**Judgment program**”, and a new calculation
239 module - **Main calculation module 2** will be used.



240
241 **Fig. 8.** Flowchart of DTORC system.

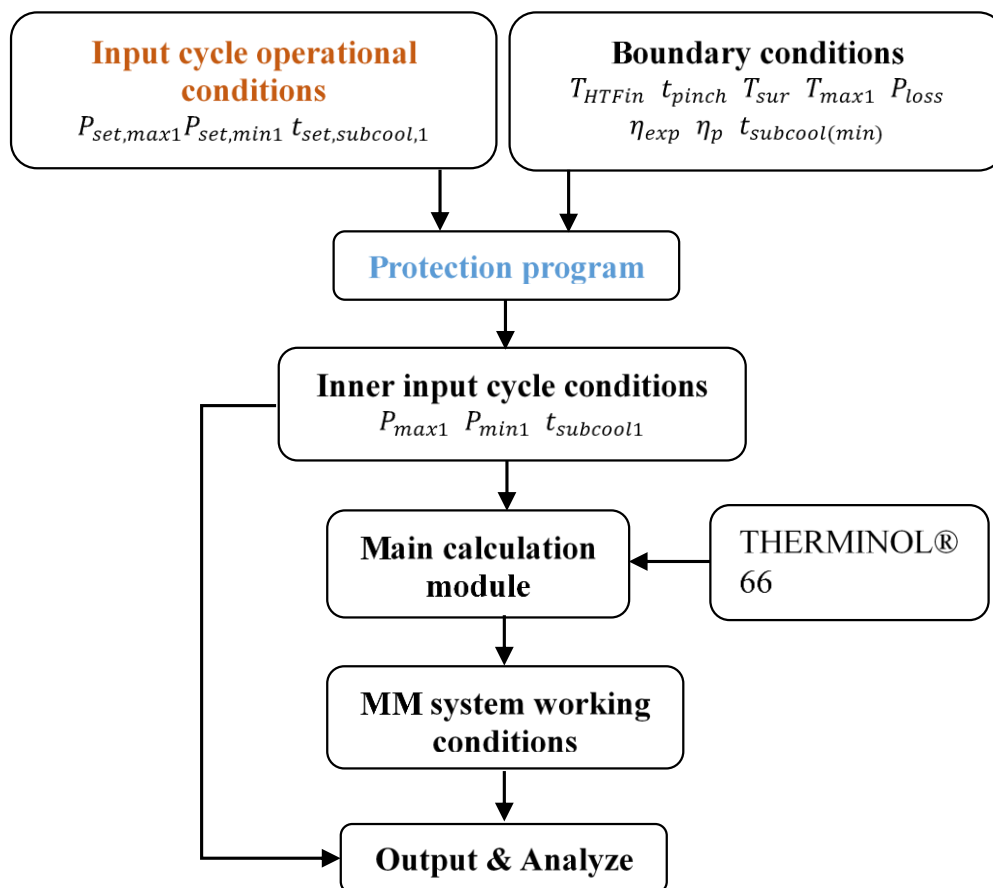
242 *3.2.3. Formulas of DTORC*

243 The main formulas used in the simulation are based on enthalpy and entropy changes.
 244 Pressure and temperature codetermine the enthalpy of a specific point except in the vaporization
 245 process and liquefaction process. There is a one-to-one correspondence between saturation
 246 temperature and saturation pressure. The enthalpy of the HTF is correlated with temperature in

247 terms of the specification supported by the manufacturer. More details can be obtained from the
 248 Appendix.

249 3.3. RC system modeling

250 It is mentioned above that T_{HTFin} is around 300 °C, and MM is also selected as the working
 251 fluid for the RC system. Considering the large positive inclination of the vapor saturation line of
 252 MM as demonstrated in Fig. 2, a recuperator is used to improve the thermal efficiency. The
 253 schematic drawing of the RC system using MM is presented in Fig. 4. The main program
 254 flowchart of the RC system is illustrated in Fig. 9. For the purpose of comparison, the RC system
 255 runs under the same heat source as the DTORC system. The main formula of the RC system is
 256 similar to that of the low temperature stage of the DTORC system in Fig. 6(b).



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Fig. 9. Flowchart of RC system.

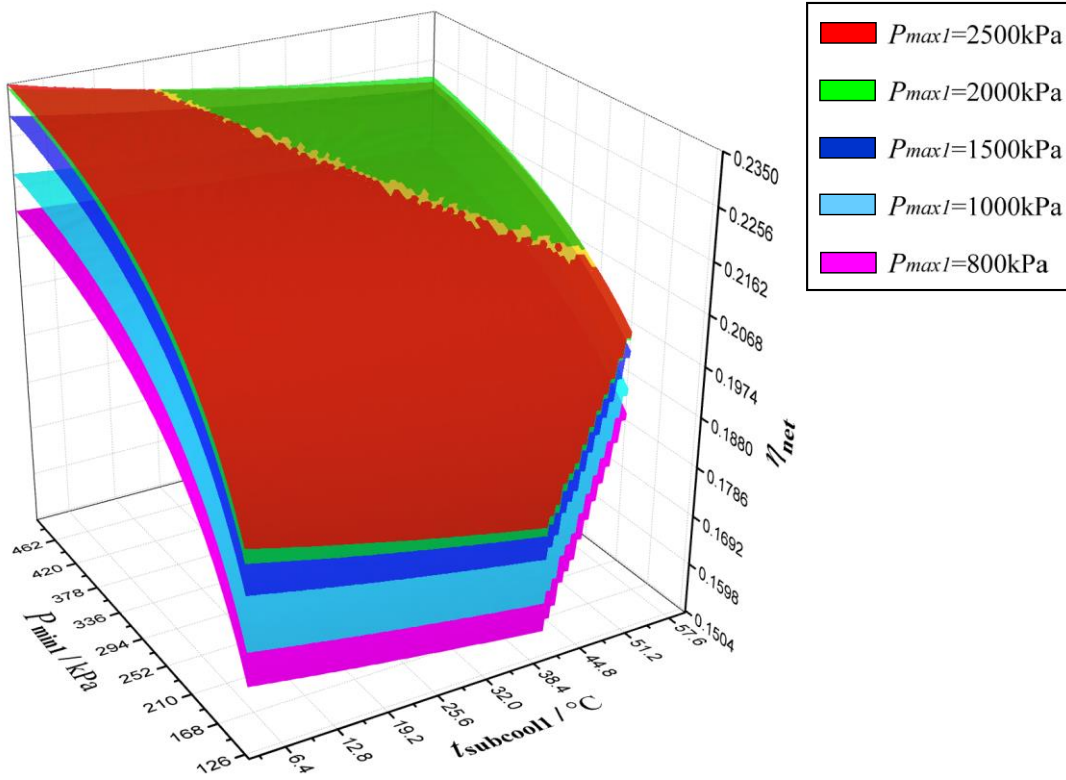
259 4. Results and discussion

260 In the current study, the overall net power output is fixed at 200 kW and the temperature of
261 the heat source is 300 °C. By comparing η_{net} , $\eta_{utilization}$ and Sur_{total} , the performance analysis
262 of DTORC is achieved. In order to analyze the effect of the working fluids on the DTORC
263 system, different power outputs at two stages are shown in the following sections.

264 4.1. Thermal efficiency and heat source utilization

265 Unlike exergy efficiency, the heat source utilization mainly focuses on the
266 entropy generation caused by the temperature difference. When the WHC is discharged into the
267 environment at a high residual temperature which is not completely utilized, the thermal
268 efficiency and exergy efficiency may still be quite high but $\eta_{utilization}$ is low. It will take the
269 whole heat source into account and higher $\eta_{utilization}$ means lower WHC consumption under the
270 same power output.

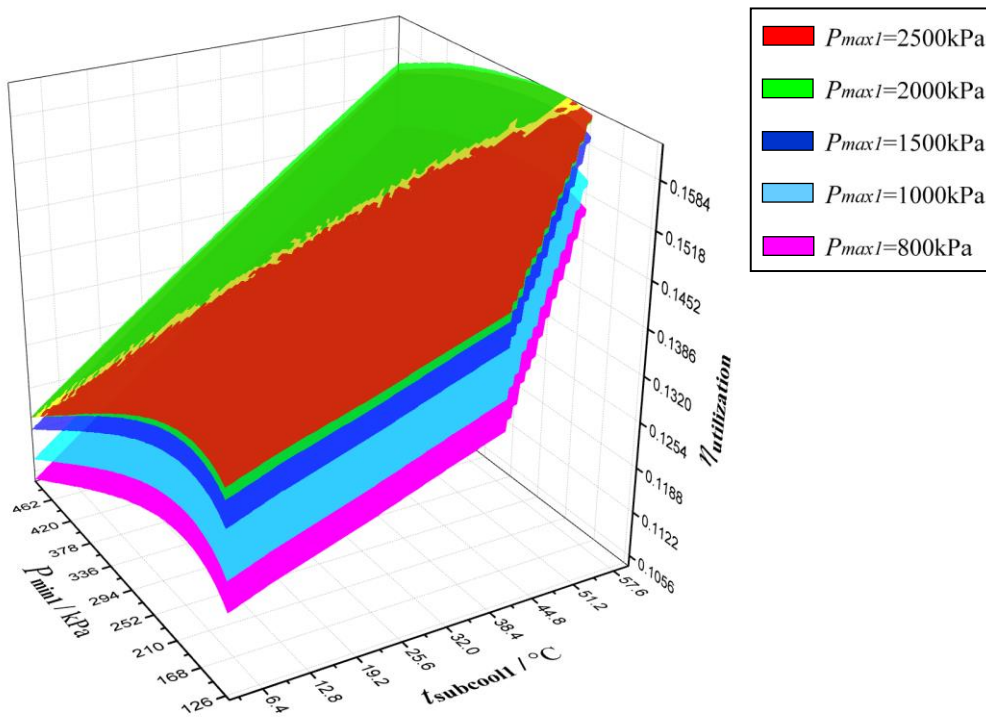
271 In the DTORC system, P_{max2} and P_{min2} are set to be maximum and minimum respectively
272 which can make the low temperature stage achieve maximum thermal efficiency at a certain heat
273 source. The heat source of the low temperature stage is determined by P_{min1} of the high
274 temperature stage. The condensation temperature of the low temperature stage is determined
275 by T_{sur} . P_{max1} , P_{min1} and $t_{subcool1}$ are changed to get the general rule of η_{net} and $\eta_{utilization}$
276 of the DTORC.



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Fig. 10. Net thermal efficiency of DTORC under different P_{max1} .



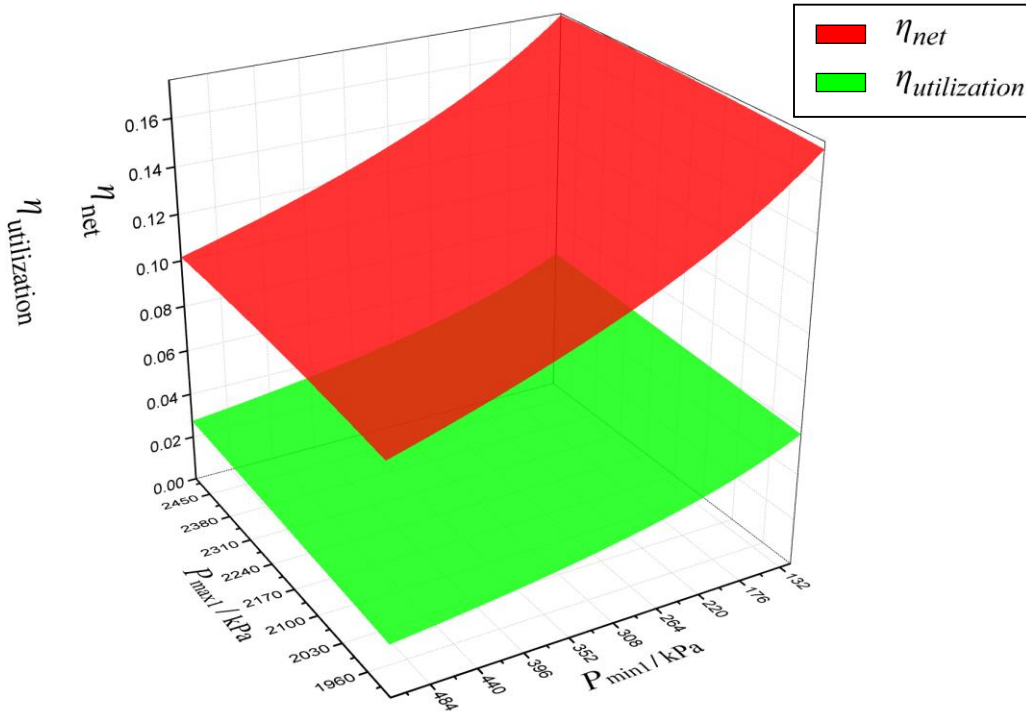
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Fig. 11. Heat source utilization of DTORC under different P_{max1} .

281 Fig. 10 and Fig. 11 show the influences of P_{min1} and $t_{subcool1}$ on η_{net} and $\eta_{utilization}$ under
282 different P_{max1} . It is found that both η_{net} and $\eta_{utilization}$ increase with the increase of P_{max1} , but
283 when P_{max1} exceeds 2000 kPa, η_{net} and $\eta_{utilization}$ decrease in some areas. When P_{max1} is
284 lower than P_c (1939.4 kPa), the addition of P_{max1} can increase the mean evaporation temperature
285 of the high temperature stage which leads to a gain of η_{net} . When P_{max1} exceeds P_c , the increase
286 of P_{max1} has a weak effect on η_{net} , and the pump power consumption becomes more and more
287 visible. The higher the P_{max1} , the more pump power it consumes. The increase of η_{net} caused by
288 the increased P_{max1} will be offset by the reduction of η_{net} caused by the pump and thus lead to a
289 reduced η_{net} in some areas, as shown Fig.10. The reduction of $t_{subcool1}$ leads to an increase
290 of η_{net} . This can be explained by the fact that the heat of the subcooled process (1-2) (see Fig. 3)
291 conducts itself to the low temperature stage. As there is pinch point temperature difference in
292 heat transfer, this means a quality reduction of the thermal energy for the low temperature stage,
293 and this will certainly result in a reduction of η_{net} . It is clear that the higher $t_{subcool,1}$ is the
294 lower η_{net} would be. But this can in some way increase $\eta_{utilization}$ as it reduces T_{HTFout} and the
295 mass flow rate of WHC. Meanwhile, the rise of P_{min1} increases the mean evaporation
296 temperature of the low temperature stage which will lead to an increased η_{net} , and it also
297 increases the mean condensation temperature at high temperature stage which will lead to a
298 reduced η_{net} . But the second stage contributes much more to η_{net} than the first stage, and the
299 comprehensive result is an increase of η_{net} . Although the increase of $t_{subcool,1}$ reduces η_{net} and
300 leads to an increase of Q_{absorb} , it will also lead to a reduction of heat source consumption thus
301 resulting in an increase of $\eta_{utilization}$, as shown in Fig.11. With $\eta_{utilization}$ up to its maximum
302 value of 16.3%, DTORC could obtain a η_{net} of 20.2% when P_{max1} is about 2000 kPa, P_{min1} 239

303 kPa and $t_{subcool1}$ 60 °C. Meanwhile, the highest temperature of the low temperature stage is
 304 228°C. The maximum η_{net} of DTORC reaches up to 23.5%.



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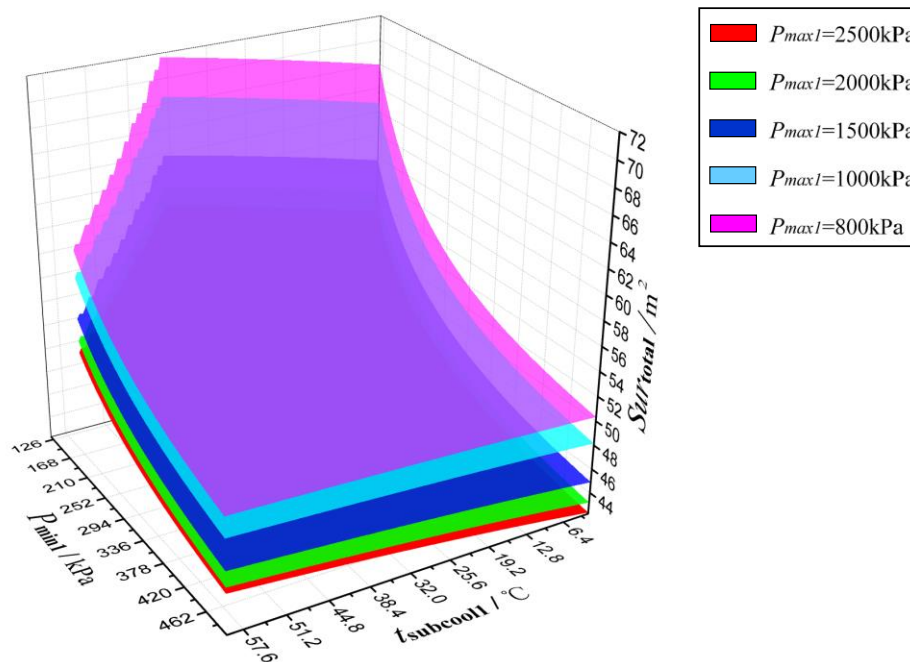
306 **Fig. 12.** Net thermal efficiency and heat source utilization of RC.

307 Fig. 12 shows $\eta_{utilization}$ and η_{net} of the RC system. Unlike the DTORC system,
 308 $\eta_{utilization}$ and η_{net} of the RC system decline with the increasing of P_{min1} . It shows that
 309 $\eta_{utilization}$ and η_{net} have the same trend, but η_{net} is more sensitive to P_{min1} than $\eta_{utilization}$. For
 310 a certain power output in this RC system, the higher η_{net} is the less \dot{m}_{HTF} would be consumed.
 311 And this will result in a high $\eta_{utilization}$. The drop of P_{min1} reduces the mean condensation
 312 temperature, leading to the increase of η_{net} . The drop of P_{min1} also reduces the outlet
 313 temperature of the expander which will cause a decline of inlet temperature of the evaporator.
 314 These lead to a reduction of \dot{m}_{HTF} and will improve $\eta_{utilization}$ (see Eq. 3). As P_{max1} has
 315 already exceeded the P_C , as shown in Fig. 12, the increase of P_{max1} will barely contribute to η_{net} .
 316 The maximum $\eta_{utilization}$ and η_{net} of RC are up to 6.5% and 18%, respectively.

317 As a consequence, under the same heat source and positive condensation pressure, the η_{net}
 318 of DTORC has a maximum increase of 30.5% greater than RC's. And $\eta_{utilization}$ of the DTORC
 319 system is almost two and a half times of RC system. In other words, the use of DTORC could
 320 increase the power output by 150% under the same mass flow rate of WHC.

321 *4.2. Total surface area of heat exchangers (Sur_{total})*

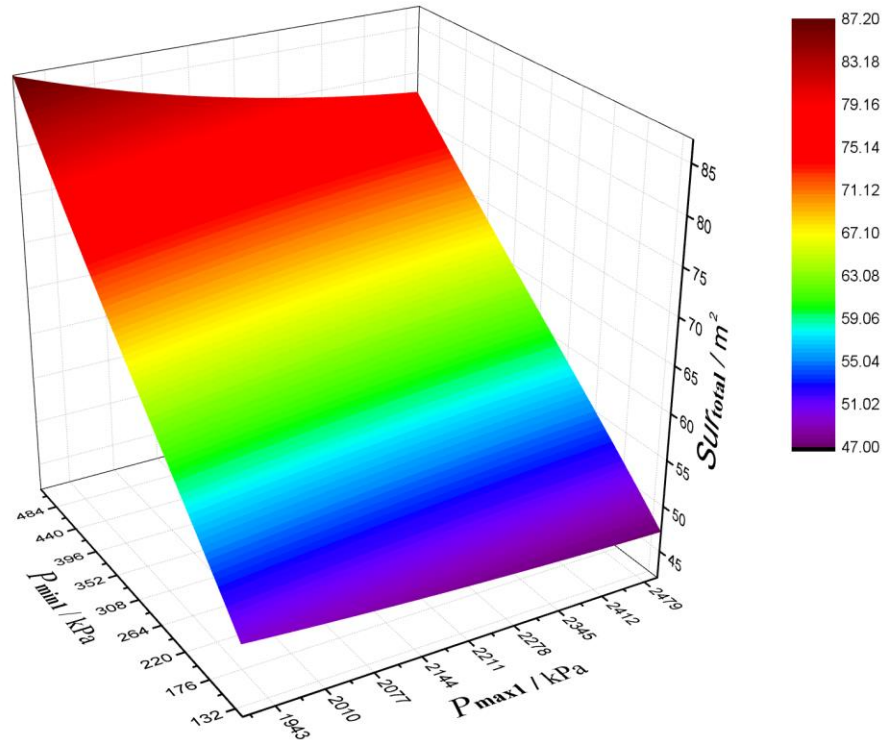
322 For a certain heat exchanger, the area needed is related to the mean temperature difference,
 323 surface heat transfer coefficient and total amount of heat transfer. Based on assumption (iii), the
 324 temperature difference is assumed to be t_{pinch} and the heat transfer coefficient is 3500 W/m²K.
 325 In the DTORC, Sur_{total} equals the evaporator and condenser surface areas of the MM cycle plus
 326 the condenser and recuperator (if any) surface areas of the R245fa cycle. The condenser of the
 327 MM cycle is also the evaporator of the R245fa cycle. The influence of P_{min1} and $t_{subcool,1}$ on
 328 the Sur_{total} of DTORC at different P_{max1} is shown in Fig. 13. The influence of P_{min1} and
 329 P_{max1} on the Sur_{total} of RC is shown in Fig. 14.



330

331

Fig. 13. Sur_{total} of DTORC under different P_{max1} .



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Fig. 14. Sur_{total} of RC with MM as the working fluid.

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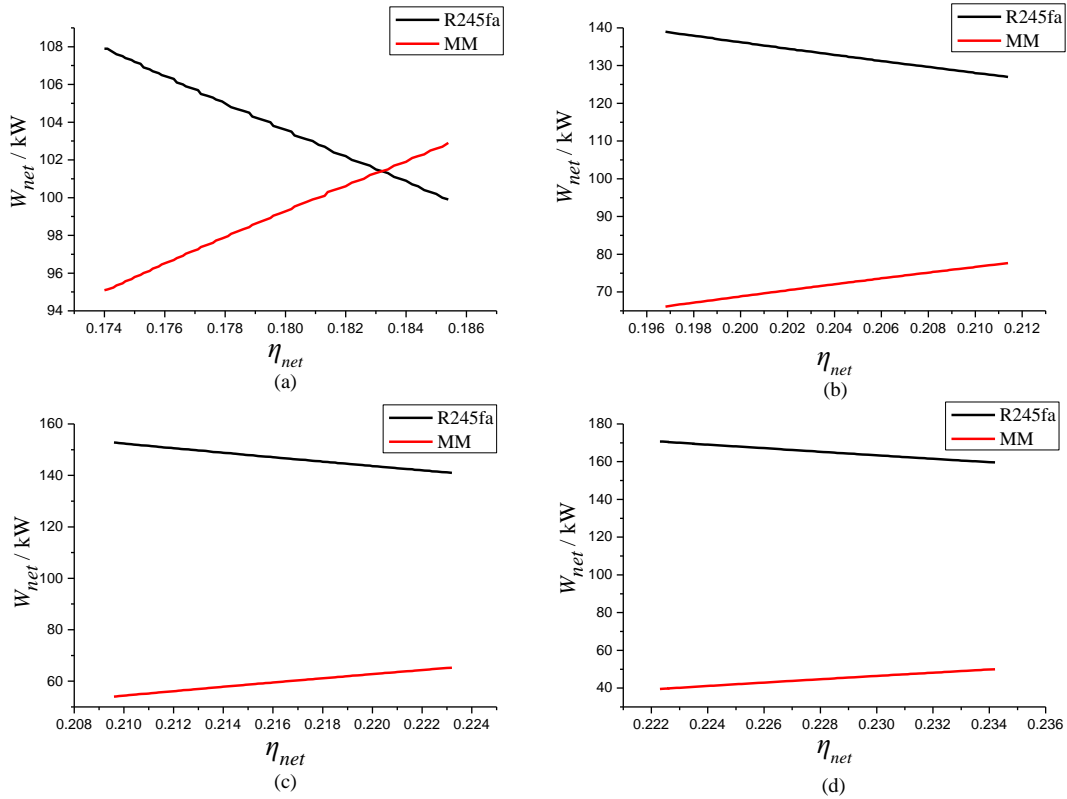
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It is found that Sur_{total} and η_{net} are changed with an opposite trend both in RC and DTORC. For a fixed power output, the higher η_{net} is the lower Q_{ab} and Q_{dis} would be, which will lead to a reduction of Sur_{total} . It is also observed that when the optimal $\eta_{utilization}$ is reached, Sur_{total} is 47 m^2 in RC and 51 m^2 in DTORC. However, the minimum Sur_{total} of DTORC is 42 m^2 and that of RC is 47 m^2 . The DTORC system has a higher $\eta_{utilization}$ and η_{net} than RC under the minimum Sur_{total} .

4.3. Respective power outputs of MM and R245fa in the DTORC



341

342 **Fig. 15.** Respective W_{net} of MM and R245fa in DTORC under $P_{max1} = 2000\text{kPa}$: (a) $P_{min1} =$
 343 120kPa ; (b) $P_{min1} = 216\text{kPa}$; (c) $P_{min1} = 308.1\text{kPa}$; (d) $P_{min1} = 500\text{kPa}$.

344 As shown in Fig. 15, it should be noted that $W_{net,MM}$ plus $W_{net,R245fa}$ equals 200 kW. In
 345 the DTORC system, R245fa obviously has a better work ability than MM. Therefore, with the
 346 increase of η_{net} , R245fa outputs more power and T_{max2} becomes higher and higher which may
 347 exceed the thermal decomposition temperature of R245fa. This should be avoided in the real
 348 applications. It should also be noted that with different combinations of organic fluids the trend
 349 of respective power outputs of the two stages may show different behaviors, due to the properties
 350 of the two working fluids. And the combination of the two working fluids could overcome the
 351 drawbacks of a single working medium.

352 **5. Conclusions**

353 A new deep super-cooling two-stage organic Rankine cycle (DTORC) is introduced to
354 enhance both the thermal efficiency and heat source utilization at high temperature waste heat
355 recovery. The performance of a DTORC system using MM and R245fa as the working fluids at
356 high and low temperature stage is investigated in details. For the purpose of comparison, a RC
357 system which uses MM as the working fluid under the same heat source at temperature of 300 °C
358 is also studied. The optimal net thermal efficiency of DTORC can be up to 23.5%, whereas it can
359 be up to 18% for RC. The optimal heat source utilization of DTORC is 16.3% whereas RC is
360 6.5%. The minimum Sur_{total} is 42 m² in DTORC and 47 m² in RC. Comparative analysis
361 between DTORC and RC may lead to the following main conclusions:

362 •A new evaluation index, effective heat source utilization, is defined and it takes the whole
363 heat source into account, which represents the working capacity of a thermal power system. High
364 heat source utilization could have high power output under the same waste heat source.

365 •The DTORC system has higher heat source utilization, approximately 2.5 times of RC. The
366 use of DTORC can increase the power output by 150% under the same mass flow rate of WHC.

367 •The maximum net thermal efficiency of DTORC could be up to 23.5%, which is increased
368 by 30.5% at high temperature waste heat recovery applications compared with that in RC.

369 •The total surface area of heat exchangers in DTORC and RC are nearly equal. The
370 minimum Sur_{total} of DTORC system is reduced by 10.6% compared with that in RC system.

371

372

373 **Appendix**

374 **In Fig. 6(a):**

375 $P_2 = P_{max1} = P_3 = P_4 + P_{loss} = P_5 + P_{loss}$

376 $P_6 = P_7 = P_8 + P_{loss} = P_1 + P_{loss} = P_{min1} + P_{loss}$

$$\begin{aligned}
377 \quad P_{10} &= P_{max2} = P_{11} = P_{12} + P_{loss} = P_{13} + P_{loss} \\
378 \quad P_{14} &= P_{15} = P_{16} + P_{loss} = P_9 + P_{loss} = P_{min1} + P_{loss} \\
379 \quad T_{HTFin} &= T_{20} = T_4 + t_{pinch} \\
380 \quad T_{HTFout} &= T_{21} = T_3 + t_{pinch} = T_2 + t_{pinch} \\
381 \quad (h_{20} - h_{21})\dot{m}_{HTF} &= (h_4 - h_3)\dot{m}_{MM} \\
382 \quad \frac{(h_{2s} - h_1)}{(h_2 - h_1)} &= \eta_p, h_1 = h_8, W_{p,MM} = (h_2 - h_1)\dot{m}_{MM} \\
383 \quad \frac{(h_5 - h_6)}{(h_5 - h_{6s})} &= \eta_{exp}, h_6 = h_7, W_{exp,MM} = (h_5 - h_6)\dot{m}_{MM} \\
384 \quad \frac{(h_{10s} - h_9)}{(h_{10} - h_9)} &= \eta_p, h_9 = h_{16}, W_{p,R245fa} = (h_{10} - h_9)\dot{m}_{R245fa} \\
385 \quad \frac{(h_{13} - h_{14})}{(h_{13} - h_{14s})} &= \eta_{exp}, h_{14} = h_{15}, W_{exp,R245fa} = (h_{13} - h_{14})\dot{m}_{R245fa} \\
386 \quad (h_7 - h_8)\dot{m}_{MM} &= (h_{12} - h_{11})\dot{m}_{R245fa} \\
387 \quad (h_{20} - h_{21})\dot{m}_{HTF} &= (h_4 - h_3)\dot{m}_{MM} \\
388 \quad Q_{ab,MM} &= (h_4 - h_3)\dot{m}_{MM}, Q_{ab,R245fa} = (h_{12} - h_{11})\dot{m}_{R245fa} \\
389 \quad Q_{dis,R245fa} &= (h_{15} - h_{16})\dot{m}_{R245fa} \\
390 \quad T_8 &= T_{11} + t_{pinch} = T_{10} + t_{pinch} \\
391 \quad T_7 &= T_{12} + t_{pinch} = T_{13} + t_{pinch} \\
392 \quad T_{16} &= T_9 = T_{sur} + t_{pinch} = T_{22} + t_{pinch} \\
393 \quad W_{net,MM} &= W_{exp,MM} - W_{p,MM}, W_{net,R245fa} = W_{exp,R245fa} - W_{p,R245fa} \\
394 \quad W_{net,sys} &= W_{net,MM} + W_{net,R245fa} \\
395 \quad \eta_{net} &= W_{net,sys}/Q_{ab,MM} \\
396 \quad \eta_{utilization} &= W_{net,sys}/((h_{20} - h_{HTF,Tsur}) * \dot{m}_{HTF})
\end{aligned}$$

$$397 \quad Sur_{total} = (Q_{ab,MM} + Q_{ab,R245fa} + Q_{dis,R245fa}) / (3500 * t_{pinch})$$

398 **In Fig. 6(b):**

$$399 \quad P_2 = P_{max1} = P_3 = P_4 + P_{loss} = P_5 + P_{loss}$$

$$400 \quad P_6 = P_7 = P_8 + P_{loss} = P_1 + P_{loss} = P_{min1} + P_{loss}$$

$$401 \quad P_{10} = P_{max2} = P_{11} = P_{14} + P_{loss} = P_{15} + P_{loss}$$

$$402 \quad P_{16} = P_{17} = P_{19} + P_{loss} = P_9 + P_{loss} = P_{min1} + P_{loss}$$

$$403 \quad T_{HTF_{in}} = T_{20} = T_4 + t_{pinch}$$

$$404 \quad T_{HTF_{out}} = T_{21} = T_3 + t_{pinch} = T_2 + t_{pinch}$$

$$405 \quad \frac{(h_{2s} - h_1)}{(h_2 - h_1)} = \eta_p, h_1 = h_8, W_{p,MM} = (h_2 - h_1)\dot{m}_{MM}$$

$$406 \quad \frac{(h_5 - h_6)}{(h_5 - h_{6s})} = \eta_{exp}, h_6 = h_7, W_{exp,MM} = (h_5 - h_6)\dot{m}_{MM}$$

$$407 \quad \frac{(h_{10s} - h_9)}{(h_{10} - h_9)} = \eta_p, h_9 = h_{19}, W_{p,R245fa} = (h_{10} - h_9)\dot{m}_{R245fa}$$

$$408 \quad \frac{(h_{15} - h_{16})}{(h_{15} - h_{16s})} = \eta_{exp}, h_{16} = h_{17}, W_{exp,R245fa} = (h_{15} - h_{16})\dot{m}_{R245fa}$$

$$409 \quad h_{12} - h_{11} = h_{17} - h_{18}$$

$$410 \quad T_{18} = T_{11} + t_{pinch}$$

$$411 \quad T_{17} = T_{12} + t_{pinch}$$

$$412 \quad T_{12} = T_{13}$$

$$413 \quad (h_7 - h_8)\dot{m}_{MM} = (h_{14} - h_{13})\dot{m}_{R245fa}$$

$$414 \quad (h_{20} - h_{21})\dot{m}_{HTF} = (h_4 - h_3)\dot{m}_{MM}$$

$$415 \quad Q_{ab,MM} = (h_4 - h_3)\dot{m}_{MM}, Q_{ab,R245fa} = (h_{14} - h_{13})\dot{m}_{R245fa}$$

$$416 \quad Q_{rec,R245fa} = (h_{17} - h_{18})\dot{m}_{R245fa}$$

417 $Q_{dis,R245fa} = (h_{18} - h_{19})\dot{m}_{R245fa}$

418 $T_8 = T_{13} + t_{pinch} = T_{12} + t_{pinch}$

419 $T_7 = T_{14} + t_{pinch} = T_{15} + t_{pinch}$

420 $T_{19} = T_9 = T_{sur} + t_{pinch} = T_{22} + t_{pinch}$

421 $W_{net,MM} = W_{exp,MM} - W_{p,MM}, W_{net,R245fa} = W_{exp,R245fa} - W_{p,R245fa}$

422 $W_{net,sys} = W_{net,MM} + W_{net,R245fa}$

423 $\eta_{net} = W_{net,sys}/Q_{ab,MM}$

424 $\eta_{utilization} = W_{net,sys}/((h_{20} - h_{HTF,T_{sur}}) * \dot{m}_{HTF})$

425 $Sur_{total} = (Q_{ab,MM} + Q_{ab,R245fa} + Q_{rec,R245fa} + Q_{dis,R245fa})/(3500 * t_{pinch})$

426

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