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**Theoretical analysis for heat exchange performance of  
transcritical nitrogen evaporator used for liquid air energy  
storage**

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## 22 Abstract

23 In view of violent changes of thermo-physical properties, the segmental design method  
24 is adopted to explore the heat exchange performances of the transcritical nitrogen (T-N<sub>2</sub>)  
25 evaporator used for liquid air energy storage, in which cold N<sub>2</sub> is heated up successively by  
26 hot propane and methanol in two wide temperature sections. The local heat capacity rate ratio  
27 between cold and hot fluids has crucial effects on the local heat exchange performance of  
28 evaporator, such as local effectiveness, local entransy dissipation, and local required heat  
29 conductance or local heat transfer rate. They have extremums near the positions where the  
30 local heat capacity rate ratio equals one, but their optimal values need to be determined by  
31 combining the changing trend of the local heat capacity rate ratio. **The total heat exchange  
32 performance of evaporator is evaluated using total entransy dissipation and total exergy  
33 efficiency. When the heat load is fixed, the total performance is improved with the decrease  
34 in the mass flow rate of methanol, but at the expense of the required total heat conductance;  
35 The total performance can be optimized by precisely tailoring the heat load ratios between the  
36 two temperature sections. When the heat conductance is given, the optimum total  
37 performance can be obtained by adjusting the mass flow rate of hot fluids at a fixed heat  
38 conductance ratio; Increasing the heat conductance ratio of the low temperature section can  
39 further elevate the optimum total performance whereas the affordable heat load or the outlet  
40 temperature of N<sub>2</sub> is notably decreased. Increasing N<sub>2</sub> pressure elevates the total performance  
41 of evaporator but diminishes the extractable cold amount from the liquid N<sub>2</sub> in the same  
42 temperature rise.** This work is beneficial for selection of key parameters to achieve optimal  
43 operation of the T-N<sub>2</sub> evaporator.

44

45 *Keywords:* Heat exchanger; Supercritical nitrogen; Entransy dissipation; Exergy efficiency;  
46 Energy storage.

## Nomenclature

### Roman letters

$c_p$	specific heat ( $\text{J}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$ )
$D_{re}$	relative difference
$\dot{E}_{dis}$	entransy dissipation ( $\text{W}\cdot\text{K}$ )
$EX$	flow exergy (W)
$h$	specific enthalpy ( $\text{J}\cdot\text{kg}^{-1}$ )
$HA$	heat conductance ( $\text{W}\cdot\text{K}^{-1}$ )
$m$	mass flow rate ( $\text{kg}\cdot\text{s}^{-1}$ )
$mc_p$	heat capacity flow rate ( $\text{W}\cdot\text{K}^{-1}$ )
$M$	number of sub-heat exchangers in the low temperature section
$N$	number of all sub-heat exchangers
$Ntu$	number of heat transfer units
$P$	pressure (Pa)
$q$	local heat transfer rate (W)
$Q_{tot}$	total heat load (W)
$R_c$	the ratio between the smaller and bigger heat capacity flow rates
$R_{c,hc}$	the ratio of heat capacity rate of hot fluid to that of cold fluid
$s$	specific entropy ( $\text{J}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$ )
$T$	temperature (K)

### Greek letters

$\varepsilon$	effectiveness
$\eta_{EX}$	exergy efficiency
$\tau$	the ratio of heat load in the low temperature section to that in the whole evaporator
$\varphi$	the ratio of heat conductance in the low temperature section to that in the whole evaporator

### Subscripts

0	environmental conditions
$c$	cold fluid
$h$	hot fluid
$hl$	hot fluid in the low temperature section
$hh$	hot fluid in the high temperature section
$i$	inlet
$j$	local position
$m$	mean value
$o$	outlet

## 47 **1. Introduction**

48 Liquid air energy storage (LAES) as a promising solution for grid scale energy storage  
49 has attracted much attention in recent years [1-5]. The LAES uses liquid air/nitrogen (N<sub>2</sub>) as  
50 both storage medium and working fluid for charging and discharging processes of electrical  
51 energy. During the charging process, excess or cheapest electricity drives air liquefaction and  
52 separation plants to produce liquid N<sub>2</sub> stored in cryogenic tanks at the nearly atmospheric  
53 pressure. During the discharging process, the liquid N<sub>2</sub> is first pressurized by a cryogenic  
54 pump and then heated up to expand in turbines to generate electricity. Cold thermal energy  
55 released in preheating of liquid N<sub>2</sub> during the discharging process can be captured to lessen  
56 refrigeration load of air liquefaction during the charging process. In view of time mismatch of  
57 the charging and discharging processes, the captured cold thermal energy required to be  
58 stored. Such a design of cold recycle based on cold storage in LAES significantly improves  
59 the overall system efficiency [2]. Conventionally, cold storage is implemented using packed  
60 beds of pebbles or rocks operating at nearly atmospheric pressure [6-8]. Operating experience  
61 of a 350 kW/2.5 MWh pilot plant located at the University of Birmingham manifested that  
62 the temporary cold storage using packed beds results in round trip efficiency improvement of  
63 LAES by ~50%. However, the dynamic effects in packed beds caused by thermal front  
64 propagation can lead to an undesired increase by 25% in the energy consumption of air  
65 liquefaction [9]. Therefore, it is required to design a novel high-efficiency cold storage unit.

66 Similar to sensible heat storage using liquids as medium, Li et al. [10] proposed a cold  
67 storage unit based on combination of two thermal fluids, which were used as both heat  
68 transfer fluids and cold storage mediums. Both She et al. [11] and Pen et al. [12] also adopted  
69 the same cold storage unit in their proposed novel LAES system. The reason for adopting two  
70 thermal fluids is that no single fluid can work totally in the form of its liquid state in the wide  
71 working temperature range of the liquid N<sub>2</sub> preheating process. The two fluids are propane

72 and methanol selected owing to their suitable working temperature ranges and comparatively  
73 large heat capacity [10]. A two-tank configuration was designed for each of the two fluids to  
74 recover and store cold energy, which can realize quasi-steady heat transfer in heat exchangers  
75 to overcome dynamic effects in packed beds [13]. The proposed unit can notably simplify the  
76 LAES system involving cold storage and offer more straightforward and flexible operating  
77 strategy with respect to the conventional packed beds [10]. The calculations indicated that the  
78 selected thermal fluids exhibit higher volume-based energy storage density than pebbles or  
79 concrete [10, 14]. This implies that a more compact system can be obtained by using the  
80 selected fluids as cold storage mediums.

81 The discharging pressure, namely the inlet pressure of the first stage turbine, is one of  
82 major operating parameters influencing the performance of LAES system. With the increase  
83 in the discharging pressure, the resulting specific expansion work increases while the  
84 recyclable cold amount diminishes [9]. In order to increase the output power of turbines, the  
85 liquid  $N_2$  is generally pressurized above the critical pressure of  $N_2$  before the inlet of first  
86 stage turbine [9, 13]. Thus  $N_2$  will undergo phase transition from the liquid state to the  
87 supercritical state in the liquid  $N_2$  preheating process. For convenience, this phase transition  
88 is also called evaporation similar to liquid-gas phase transition, and the corresponding heat  
89 exchanger is named transcritical  $N_2$  (T- $N_2$ ) evaporator in the present paper. The performance  
90 of the evaporator determines the amount of recovered cold and the inlet temperature of  
91 turbines in a LAES system, and thus has crucial influences on the operation efficiency and  
92 stability of the system [9]. However, the thermodynamic properties of  $N_2$  change dramatically  
93 around the pseudo-critical temperature, which makes the heat transfer in the evaporator rather  
94 complicated and the design of the evaporator very challenging.

95 Some studies have been devoted to the heat transfer characteristics of supercritical  $N_2$   
96 [15-19]. Dimitrov et al. [15] conducted experiments on forced convective heat transfer of

97 supercritical nitrogen at a pressure of 4 MPa in a vertical tube. The results indicated that the  
98 heat transfer can be enhanced when the difference between wall and bulk temperatures spans  
99 the drastic variation region of the thermo-physical properties of N<sub>2</sub>. Zhang et al. [17] carried  
100 out experimental and numerical studies on flow and heat transfer of supercritical N<sub>2</sub> in a  
101 vertical mini-tube. They reported that there is **considerable** discrepancy in Nusselt numbers  
102 between the experimental results and the predictions by the existing correlations. Ciprian et  
103 al. [19] numerically examined the heat transfer coefficient of supercritical N<sub>2</sub> in the large  
104 specific heat region flowing upward in a vertical tube under different operating pressures.  
105 They found that the increase of heat flux could cause heat transfer deterioration. The above  
106 studies almost focus on the heat transfer behaviours of supercritical N<sub>2</sub> under fixed heat flux  
107 conditions. Actually, the heat transfer in the evaporator is coupled between the N<sub>2</sub> and the  
108 transfer fluids, and hence the heat transfer condition of N<sub>2</sub> is changing along the flow  
109 direction of N<sub>2</sub>. Therefore, deep understanding of coupled the heat transfer behaviours  
110 between N<sub>2</sub> and two heat transfer fluids in the evaporator is crucial to the optimization design  
111 of evaporator for improving performances of the LAES system.

112 From the above, although such a T-N<sub>2</sub> evaporator including the combination of propane  
113 and methanol as heat transfer fluids has been adopted by several researches [10-12], studies  
114 available in the literature have not addressed the following key aspects: (1) the local and  
115 overall heat exchange performances of the evaporator coupled with three fluids in the case of  
116 drastic change of the thermo-physical properties of N<sub>2</sub>; (2) how to select the key operating  
117 parameters, including mass flow rates of heat transfer fluids, inlet pressure of N<sub>2</sub>, and heat  
118 load or conductance distribution ratio between the two heat transfer fluids. Therefore, this  
119 paper adopts the segmental design method [20] to precisely capture the coupled heat transfer  
120 behaviours in the T-N<sub>2</sub> evaporator, and employs the entransy dissipation theory [21] and  
121 exergy analysis method [22] to evaluate the heat exchange performance of the evaporator.

122 The local performances of the evaporator as well as its overall performances are explored in  
123 detail from the respective viewpoints of design and check calculations. The effects of the key  
124 operating parameters on the entransy dissipation, exergy efficiency, and required heat  
125 conductance or affordable heat load of the evaporator are examined for optimization of these  
126 parameters. This study can provide significant references for optimization of T-N<sub>2</sub> evaporator  
127 to achieve high-efficiency cold storage in the LAES system.

128

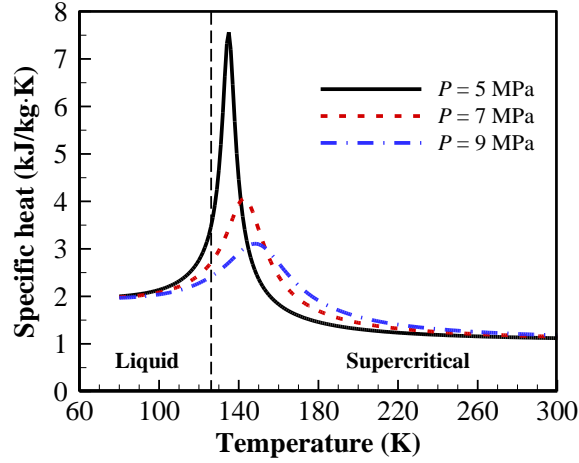
## 129 **2. Theoretical approach**

### 130 *2.1. Segment design and main assumptions*

131 The critical point of N<sub>2</sub> locates at (126.19 K, 3.4 MPa). Fig.1 shows variation of the  
132 specific heat of N<sub>2</sub> with temperature above critical pressures. It is evident that the specific  
133 heat exhibits severe change near pseudo-critical point, which is beneficial to extracting more  
134 cold energy, but makes the heat exchanger design more difficult than fixed properties.  
135 According to the respective working temperature ranges of the selected two heat transfer  
136 fluids, the evaporator is artificially divided into a low temperature section for propane and a  
137 high temperature section for methanol. The two hot fluids are used to successively heat up the  
138 cold fluid N<sub>2</sub> [10-12]. Based on the segment design method as mentioned above, the  
139 evaporator with counter-flow configuration is discretized into a sufficient number of serial  
140 sub-heat exchangers (SHEs) as depicted in Fig. 2 [20, 23], where  $T$  denotes the temperature;  
141  $q$  indicates the local heat transfer rate; subscripts  $c$  and  $h$  denote the cold and hot fluids,  
142 respectively; subscript  $j$  indicates the local position; subscripts  $hl$  and  $hh$  indicate the hot  
143 fluids in the low and high temperature sections, respectively; subscripts  $i$  and  $o$  denote inlet  
144 and outlet, respectively;  $N$  and  $M$  represent the numbers of SHEs in the whole evaporator and  
145 the low temperature section, respectively.  $M$  can be calculated by  $M = \tau N$  for design  
146 calculation or  $M = \varphi N$  for check calculation, where  $\tau$  and  $\varphi$  represent the ratios of heat load

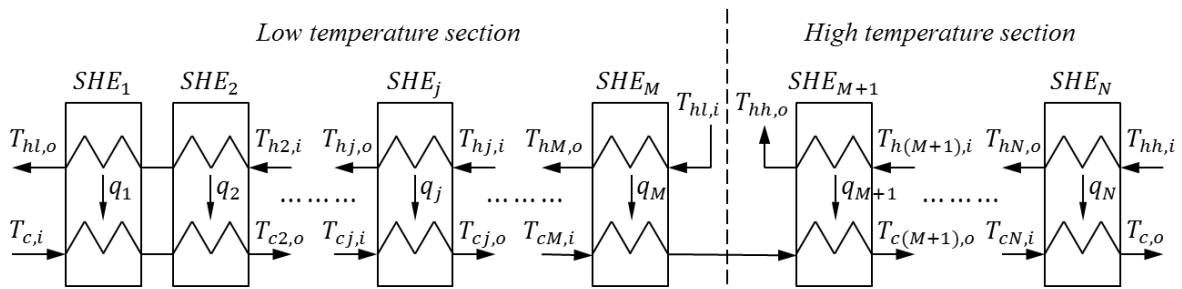


147 and heat conductance in the low temperature section to those in the whole evaporator,  
 148 respectively. An equality of temperature exists at the junction of each two SHEs for the cold  
 149 and hot fluids, respectively.



150  
 151  
 152

Fig. 1. The specific heat of N<sub>2</sub> dependent on temperature at pressures of 5 MPa, 7 MPa and 9 MPa.



153  
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 155

Fig. 2. The schematic diagram of SHEs for three types of fluids.

156 In order to formulate the heat exchange problem in a T-N<sub>2</sub> evaporator, the following  
 157 assumptions were made: (a) the specific heats of cold and hot fluids are considered to be  
 158 constant in each SHE **as the evaporator is divided into enough number of SHEs**; (b) the heat  
 159 transfer coefficient is considered to be invariable in each SHE **as the number of SHEs is**  
 160 **enough**; (c) there is no heat or cold loss from the evaporator to environment **due to good**  
 161 **thermal insulation layers on the outer surfaces of evaporator**; (d) the heat conduction along  
 162 axial direction in heat transfer surface of the evaporator is negligible **since the thickness of**  
 163 **heat transfer surface of the evaporator is quite small compared to its axial length**; (e) the  
 164 pressure drop of cold and hot fluids caused by flow friction is ignorable **considering that the**

165 pressure drop by flow friction is very small compared to the inlet pressure and has tiny  
166 influences on the thermo-physical properties of fluids; (f) for the sake of ensuring the  
167 continuity of heat exchange and avoiding heat exchange between the two hot fluids, it is  
168 assumed that the inlet temperature of the hot fluid in the low temperature section is equal to  
169 the outlet temperature of the hot fluid in the high temperature section, i.e.  $T_{hl,i} = T_{hh,o}$ . This  
170 constraint condition can easily be achieved by reasonable design; (g) a safe temperature  
171 difference of 5 K, with respect to the freezing point or boiling point of each hot fluid, is  
172 specified to ensure that each hot fluid operates at the liquid state in the evaporator. The  
173 proposed safe temperature difference of 5 K is enough for avoiding freezing or boiling of hot  
174 fluids caused by occasional temperature fluctuation, and it is not too large to markedly  
175 narrow the optional range of mass flow rates of hot fluids. As a result of the assumptions (a-  
176 d), the effectiveness—number of heat transfer unit ( $\epsilon - NTU$ ) method is applicable for each  
177 SHE [23].

178

## 179 2.2. Basic equations based on segment design method

180 For the design calculation, the total heat load of evaporator denoted by  $Q_{tot}$  is fixed,  
181 which can be determined by the given mass flow rate, inlet and outlet statuses of  $N_2$  as  
182 follows:

$$Q_{tot} = m_c(h_{c,o} - h_{c,i}), \quad (1)$$

183 where  $m$  is the mass flow rate of fluid and  $h$  is the specific enthalpy. The specific enthalpies  
184 are totally dependent on the given temperatures and pressures at the inlet and outlet of  $N_2$ .

185 It is prescribed that the total heat load is evenly divided among the SHEs, and thus the  
186 local heat transfer rate of each SHE can be calculated by  $q_j = Q_{tot}/N$ . According to the  
187 definition, the local effectiveness of each SHE can be written as [24]

$$\varepsilon_j = \frac{q_j / \min(m_c c_{pj,c}, m_h c_{pj,h})}{(T_{hj,i} - T_{cj,i})}, \quad (2)$$

188 where  $c_p$  is the specific heat of fluids.  $m c_p$  as an important variable is referred to as heat  
 189 capacity flow rate of fluids, which can be obtained for the cold and hot fluids in each SHE as  
 190 follows [23]:

$$m_c c_{pj,c} = \frac{q_j}{T_{cj,o} - T_{cj,i}}, \quad (3)$$

$$m_h c_{pj,h} = \frac{q_j}{T_{hj,i} - T_{hj,o}}. \quad (4)$$

191 In view of the one-to-one correspondence between the temperature and specific  
 192 enthalpy at a fixed pressure, the unknown end temperature of each SHE can be obtained  
 193 through the energy balance in each SHE. The energy balance in each SHE can be expressed  
 194 as

$$m_c (h_{cj,o} - h_{cj,i}) = m_h (h_{hj,i} - h_{hj,o}) = q_j. \quad (5)$$

195 Based on the  $\varepsilon - NTU$  method, the local number of heat transfer unit for each counter-  
 196 flow SHE can be written as [24]

$$NTU_j = \frac{\ln((1 - \varepsilon_j)/(1 - R_{cj}\varepsilon_j))}{R_{cj} - 1}, \quad (6)$$

197 where  $R_{cj}$  is defined as

$$R_{cj} = \frac{\min(m_c c_{pj,c}, m_h c_{pj,h})}{\max(m_c c_{pj,c}, m_h c_{pj,h})}. \quad (7)$$

198 For each SHE, the required local heat conductance can be expressed as [24]

$$HA_j = NTU_j \min(m_c c_{pj,c}, m_h c_{pj,h}). \quad (8)$$

199 Summing the local heat conductance yields the required total heat conductance in the  
 200 evaporator:

$$HA = \sum_1^N HA_j. \quad (9)$$

201 For the check calculation, the total heat conductance is fixed and each SHE has the  
 202 same local heat conductance. In this circumstance, the local number of heat transfer unit in  
 203 each SHE can be obtained by

$$Ntu_j = \frac{HA_j}{\min(m_c c_{p,j,c}, m_h c_{p,j,h})}. \quad (10)$$

204 The local effectiveness in each SHE can be expressed as

$$\varepsilon_j = \frac{1 - \exp(-NTU_j(R_{c,j} - 1))}{1 - R_{c,j}\exp(-NTU_j(R_{c,j} - 1))}. \quad (11)$$

205 The affordable local heat load of each SHE can be calculated by

$$q_j = \varepsilon_j \min(m_c c_{p,j,c}, m_h c_{p,j,h})(T_{h,j,i} - T_{c,j,i}). \quad (12)$$

206 The affordable total heat load of the evaporator can be obtained by summing the local  
 207 heat load. According to the  $\varepsilon - NTU$  method, the unknown end temperatures of cold and hot  
 208 fluids in each SHE can be obtained by iteration based on the energy balance in Eq. (5).

209 The entransy dissipation introduced by Guo et al. [21, 25] can be used to reflect the  
 210 irreversibility of a heat transfer process. The entransy dissipation has been successfully  
 211 employed to evaluate or optimize heat transfer performance in some recent studies [26-29].  
 212 Based on the assumption (c), the local entransy dissipation in each SHE can be expressed as  
 213 [30-32]

$$\dot{E}_{dis,j} = \frac{1}{2}(m_h c_{p,j,h} T_{h,j,i}^2 - m_h c_{p,j,h} T_{h,j,o}^2) + \frac{1}{2}(m_c c_{p,j,c} T_{c,j,i}^2 - m_c c_{p,j,c} T_{c,j,o}^2). \quad (13)$$

214 The total entransy dissipation in the evaporator can be written as

$$\dot{E}_{dis} = \sum_1^N \dot{E}_{dis,j}. \quad (14)$$

215 Exergy efficiency is another important indicator to the performance of heat exchanger,  
 216 which reveals the quality of the usable energy transfer [33-35]. Since the working  
 217 temperature of fluids in the evaporator is below the environmental temperature (assumed to  
 218 be 293 K), cold exergy is transferred from the cold fluid to the two hot fluids. The total  
 219 exergy efficiency of the evaporator is defined as

$$\eta_{EX} = \frac{EX_{hl,o} - EX_{hl,i} + EX_{hh,o} - EX_{hh,i}}{EX_{c,i} - EX_{c,o}}. \quad (15)$$

220 The term  $EX$  represents the flow exergy of each fluid at the respective inlet or outlet  
 221 in the evaporator, which can be calculated as follows [36, 37]:

$$EX = m[(h - h_0) - T_0(s - s_0)], \quad (16)$$

222 where  $s$  is the specific entropy of fluid and the subscript 0 denotes the environmental  
 223 conditions.

224 By substituting Eq. (16) into Eq. (15), the exergy efficiency can be expressed as

$$\eta_{EX} = \frac{m_{hl}[(h_{hl,o} - h_{hl,i}) - T_0(s_{hl,o} - s_{hl,i})] + m_{hh}[(h_{hh,o} - h_{hh,i}) - T_0(s_{hh,o} - s_{hh,i})]}{m_c[(h_{c,i} - h_{c,o}) - T_0(s_{c,i} - s_{c,o})]}, \quad (17)$$

225 which combined with the heat balance relation  $m_{hl}(h_{hl,i} - h_{hl,o}) + m_{hh}(h_{hh,i} - h_{hh,o}) =$   
 226  $Q_{tot}$  and Eq. (1) yields

$$\eta_{EX} = \frac{T_0[m_{hl}(s_{hl,i} - s_{hl,o}) + m_{hh}(s_{hh,i} - s_{hh,o})] - Q_{tot}}{T_0 m_c (s_{c,o} - s_{c,i}) - Q_{tot}}. \quad (18)$$

227

### 228 2.3. Design parameters and calculation procedures

229 The initial given parameters for both design and check calculations are listed in Table 1.

230 The liquid air is generally stored in a tank at a temperature of about 80 K. When the liquid air  
 231 is pressurized by a cryogenic pump for flowing into the evaporator to release cold energy, its  
 232 temperature is generally increased to about 83 K [9, 10]. Thus, the inlet temperature of cold

233 fluid is set to 83 K. The inlet pressure of cold fluid is set to 5 MPa, 7 MPa and 9 MPa to  
 234 explore its effects on heat exchange performance of the evaporator. The inlet pressures of hot  
 235 fluids are set to atmospheric pressure, i.e. 0.1 MPa. As mentioned in Section 2.1, propane and  
 236 methanol are selected as hot fluids to successively heat up the cold fluid N<sub>2</sub>. The temperature-  
 237 dependent specific enthalpies and entropies of N<sub>2</sub>, propane and methanol at prescribed  
 238 pressures were extracted from NIST standard database [38]. The temperature-dependent  
 239 specific enthalpies are also depicted in Fig. 3. Based on the assumption (e), it is confirmed  
 240 that these isobaric specific enthalpies can be used to support the calculations for the whole  
 241 evaporator at a specified inlet pressure. Fig. 3(a) shows that the specific enthalpy of N<sub>2</sub> at a  
 242 fixed pressure varies nonlinearly with the temperature. According to the freezing points and  
 243 boiling points of the hot fluids at the pressure of 0.1 MPa as shown in Fig. 3(b) combined  
 244 with the assumption (g), it can be deduced that the operating temperature of propane and  
 245 methanol should be maintained between 90.5 K and 225.7 K and between 180.6 K and 332.3  
 246 K, respectively. In light of the assumption (f), it can be further obtained that the inlet  
 247 temperature of propane or the outlet temperature of methanol should be maintained between  
 248 180.6 K and 225.7 K for reliable operation, as shown in Fig. 3(b).

249

250 Table 1 The initial given data for design and check calculation.

Parameters	Values for design calculation	Values for check calculation
Inlet pressure of cold fluid, $P_c$ (MPa)	5, 7, 9	5, 7, 9
Inlet temperature of cold fluid, $T_{c,i}$ (K)	83	83
Outlet temperature of hot fluid, $T_{c,o}$ (K)	283	-
Heat conductance, HA (MW/K)	-	4
Mass flow rate of cold fluid, $m_c$ (kg/s)	100	100
Inlet pressure of hot fluids, $P_h$ (MPa)	0.1	0.1
Inlet temperature of hot fluid at high temperature section, $T_{hh,i}$ (K)	288	288
Outlet temperature of hot fluid at low temperature section, $T_{hl,o}$ (K)	93	93
Number of SHEs, $N$	80	80

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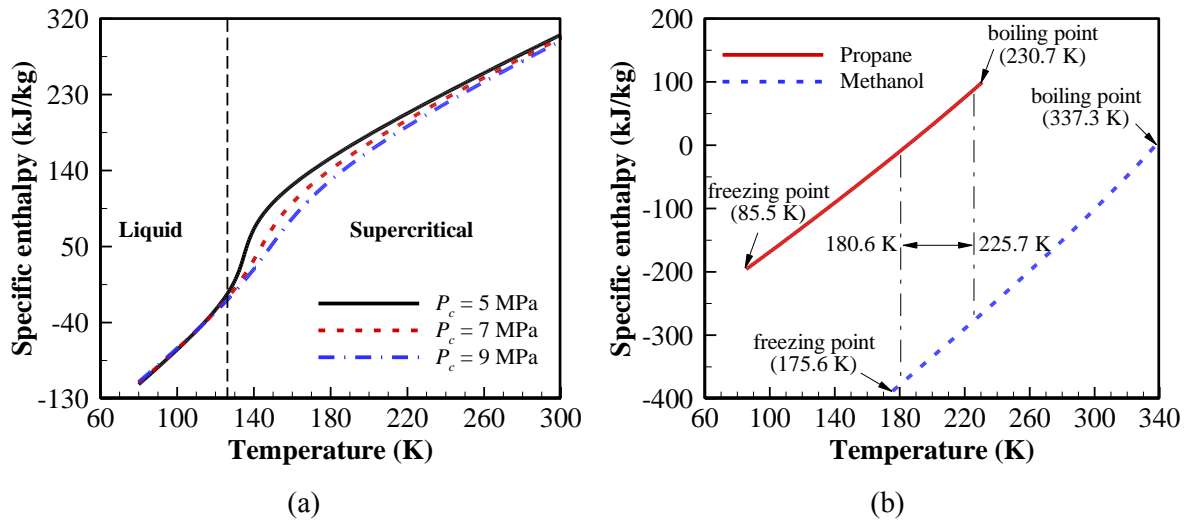


Fig. 3. The temperature-dependent specific enthalpies at specified pressures: (a)  $N_2$  at  $P_c=5$  MPa,  $P_c=7$  MPa, and  $P_c=9$  MPa; and (b) propane and methanol at  $P_h=0.1$  MPa.

Fig. 4(a) illustrates the calculation procedures for design calculation, which includes the input and output parameters. The input parameters,  $m_{hh}$  and  $\tau$ , are not the initial design parameters and thus not specified in Table 1. They are variable and need to be given and adjusted to reveal their effects on heat exchange performance. The critical parameter  $m_{hl}$  is initially supposed to start the iterations. Subsequently, the local parameters are successively calculated for each SHE starting from the outlet of cold fluid, except for the first SHE. The  $m_{hl}$  is then refreshed based on the first SHE using Eq. (5) for the next iteration until satisfying convergence criterion. In the new iterations after the  $m_{hl}$  is refreshed, the calculations of local parameters for the high temperature section can be skipped. After the calculation convergence is achieved, some local or holistic parameters can be easily obtained, including mass flow rate and inlet temperature of hot fluid in the low temperature section, effectiveness, entransy dissipation and required heat conductance.

Fig. 4(b) depicts the calculation procedures for check calculation, which involves one more iteration loop compared to the procedures for design calculation. Similarly, the input parameters,  $m_{hl}$  and  $\varphi$ , are variable and need to be given and adjusted to reveal their effects

272 on heat exchange performance. In the iteration loop of inner layer, the supposed parameter  $q_j$   
273 for each SHE is repeatedly refreshed until convergence. In the iteration loop of middle layer,  
274 the calculation of local parameters for each SHE, except for the last SHE, starts from the inlet  
275 of cold fluid, since the outlet temperature of cold fluid is unknown. In the iteration loop of  
276 outer layer, the supposed parameter  $m_{hh}$  is repeatedly refreshed based on the last SHE until  
277 convergence. Similarly, in the new iterations after the  $m_{hh}$  is refreshed, the calculations of  
278 local parameters for the low temperature section can be skipped. After the calculation  
279 convergence is achieved, some local or holistic parameters can be easily obtained, including  
280 mass flow rate of hot fluid in the high temperature section, inlet temperature of hot fluid in  
281 the low temperature section, effectiveness, entransy dissipation and affordable heat load.

282 For simulating the heat exchange characteristics of the T-N<sub>2</sub> evaporator, two sets of  
283 programs were developed in the MATLAB software based on the above calculation  
284 procedures for the design and check calculations, respectively.



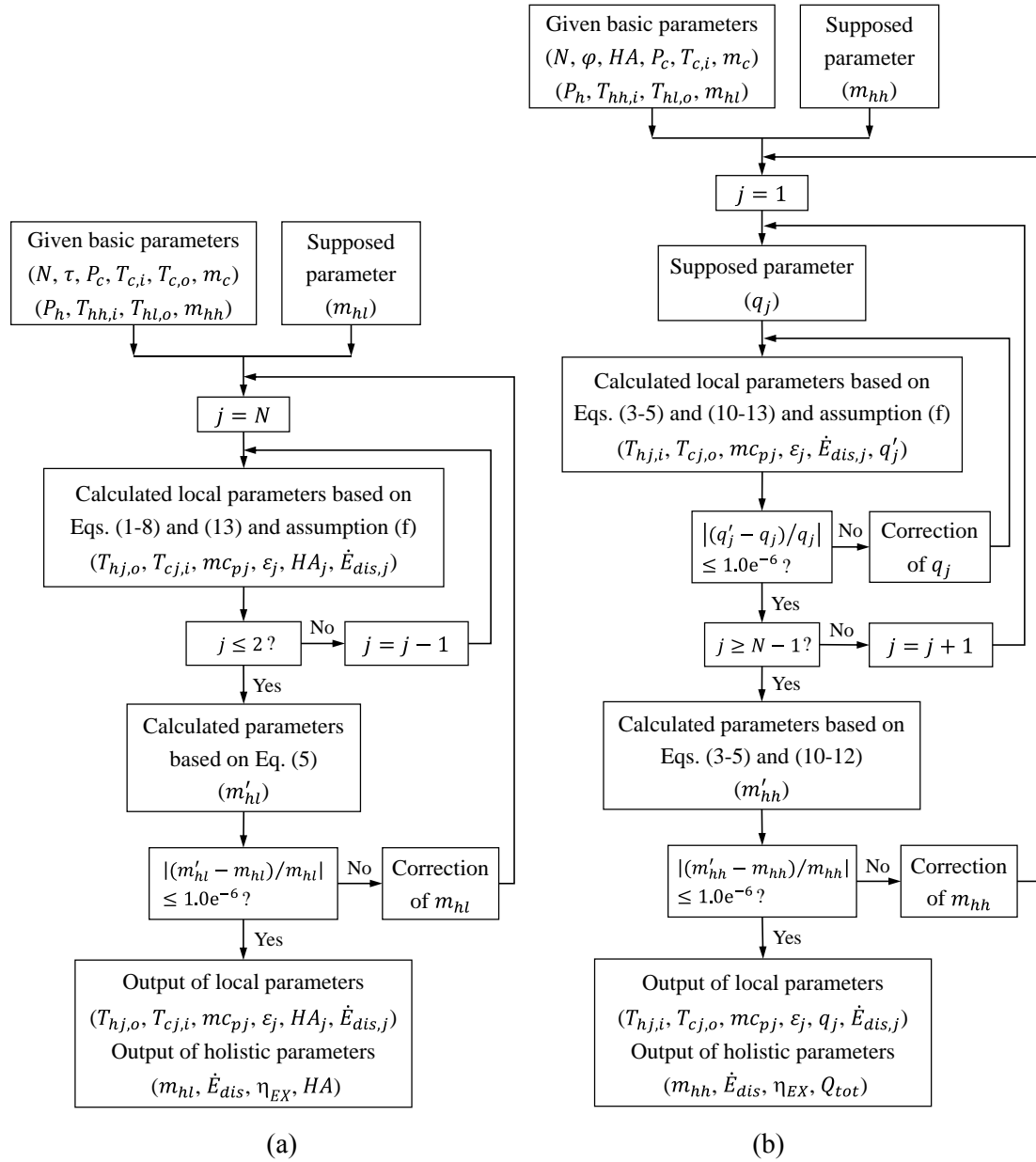


Fig. 4 Calculation procedures for (a) design calculation (b) check calculation.

#### 2.4. Segment independent study

The number of SHEs ( $N = 80$ ) as listed in Table 1 is determined by segment independent test. Four different numbers of SHEs, 20, 40, 80 and 160, were used to obtain a segment independent solution. For the segment independent test of design calculation, the heat load ratio of low temperature section was set to 65% and the mass flow of methanol was set to 60 kg/s. The predicted total entransy dissipation and total heat conductance for design

295 calculation with the four segment settings are shown in Table 2. Similarly, for the segment  
 296 independent test of check calculation, the heat conductance ratio of low temperature section  
 297 was set to 65% and the mass flow of propane was set to 120 kg/s. The predicted total  
 298 entransy dissipation and mass flow of methanol for check calculation with the four segment  
 299 settings are shown in Table 2. The relative difference between values predicted with two  
 300 adjacent segment settings is denoted by  $D_{re}$ , which is also listed in Table 2. From the table, it  
 301 is obvious that all the relative differences of the selected four parameters are less than 0.05%  
 302 between the segment settings of  $N = 80$  and  $N = 160$ . Therefore, the segment setting of  
 303  $N = 80$  is selected for both design and check calculations to complete the numerical  
 304 predictions in the following section.

306 Table 2 Segment independent results for design and check calculations.

$N$	Design calculation				Check calculation			
	$\dot{E}_{dis}(MW \cdot K)$	$D_{re}(\%)$	$HA(MW/K)$	$D_{re}(\%)$	$\dot{E}_{dis}(MW \cdot K)$	$D_{re}(\%)$	$m_{hh}(kg/s)$	$D_{re}(\%)$
20	384.53	-	4.2923	-	428.46	-	55.5819	-
40	385.64	0.29	4.2947	0.06	431.07	0.61	55.4567	0.23
80	385.92	0.07	4.2953	0.01	431.79	0.17	55.4218	0.06
160	385.98	0.02	4.2955	0.00	431.97	0.04	55.4119	0.02

### 308 3. Results and discussions

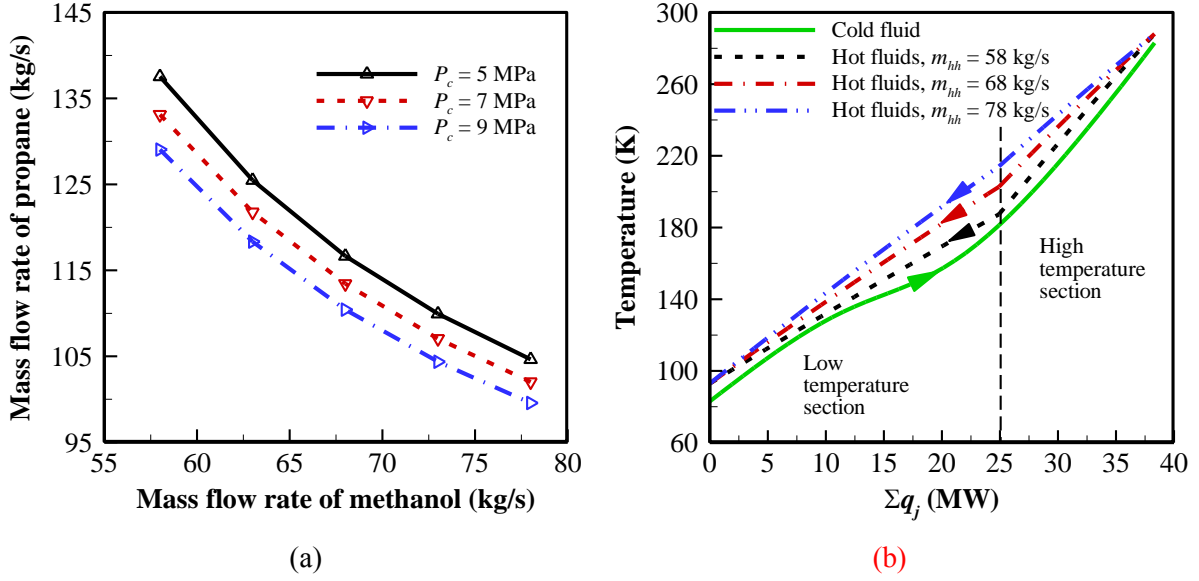
#### 309 3.1. Design calculation

310 As listed in Table 1, the inlet and outlet temperatures of cold fluid as well as its mass  
 311 flow rate are fixed, and thus the total heat load and local heat transfer rate are fixed. The inlet  
 312 temperature of hot fluid at the high temperature section and outlet temperature of hot fluid at  
 313 the low temperature section are also fixed to determine the mass flow rate of propane, under  
 314 the selected mass flow rate of methanol and the selected heat load ratio of low temperature  
 315 section based on the assumption (f). The heat load ratio of low temperature section was set to  
 316 65% as an illustration. The quantitative relations of the mass flow rates of the two hot fluids

317 under different pressures of cold fluid are shown in Fig. 5(a). The mass flow rate of propane  
318 decreases with the increase in that of methanol. Since the corresponding relation is unique at  
319 a specified pressure of cold fluid, only the mass flow rate of methanol  $m_{hh}$  is referred to in  
320 the following for convenience. Fig. 5(b) illustrates the relations of temperature with the local  
321 heat transfer rate accumulation under different mass flow rates of hot fluids. The temperature  
322 shows twisty variation along the flow direction of cold fluid in spite of same local heat  
323 transfer rate existing in each SHE, which is attributed to the varying specific heat of N<sub>2</sub>. With  
324 the decrease in  $m_{hh}$ , the pinch point in the low temperature section gradually moves from the  
325 inlet of cold fluid to the interior for the whole evaporator and the temperature difference at  
326 the pinch point also gradually decreases as indicated in Fig. 5(b). This implies that the  
327 violently varying properties of fluids easily result in the temperature cross or violating the  
328 pinch constrains, which makes the evaporator invalid. The calculation indicates that when  
329  $m_{hh} \leq 55$  kg/s the temperature cross will occur. Therefore, it is of vital importance to  
330 carefully tailor the relevant parameters in the design of T-N<sub>2</sub> heat exchanger. In addition, Fig.  
331 5 (b) indicates that the required inlet temperature of hot fluid in the low temperature section  
332 descends with the decreases in  $m_{hh}$ .

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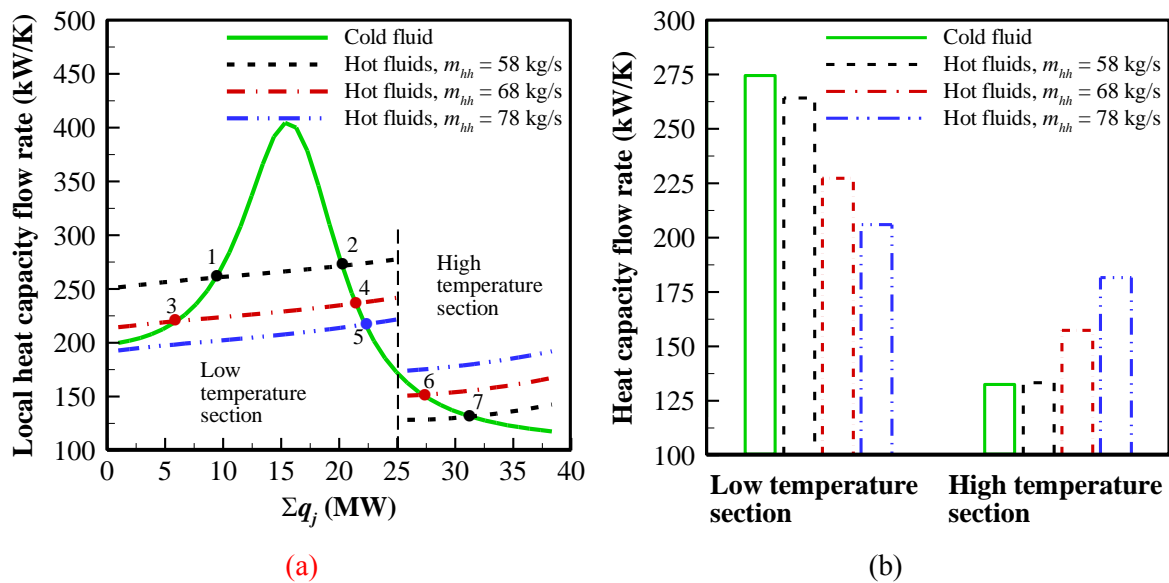
Fig. 5. The relations of (a) mass flow rates of the two hot fluids under different pressures of cold fluid at  $\tau = 65\%$ , and (b) temperature with local heat transfer rate accumulation in SHEs at  $P_c = 7$  MPa and  $\tau = 65\%$ .

341 Fig. 6(a) shows the relations of local heat capacity flow rate with the local heat transfer  
342 rate accumulation under different mass flow rates of hot fluids. Along the flow direction of  
343 cold fluid, the heat capacity flow rate of cold fluid first increases and then decrease while the  
344 heat capacity flow rates of hot fluids gradually increase in both the two sections. The  
345 inflection point of changing trend for the heat capacity flow rate of cold fluid exists at the low  
346 temperature section. The changing range of heat capacity flow rate of cold fluid is much  
347 larger than those of hot fluids. The discrepancy of the heat capacity flow rates between hot  
348 fluids in the two sections decreases with the increase in  $m_{hh}$ . The variation curves of the heat  
349 capacity flow rates of cold and hot fluids intersect with each other at some locations, and the  
350 number of intersection points reduces from 3 to 1 when  $m_{hh}$  increases from 58 kg/s to 78  
351 kg/s. The local heat capacity rate ratio is defined as

$$R_{cj,hc} = \frac{m_h c_{pj,h}}{m_c c_{pj,c}}. \quad (19)$$

352 Obviously, the local heat capacity rate ratio equals one at these intersection points as shown  
 353 in Fig. 6(a). The respective average heat capacity flow rates of cold and hot fluids in the two  
 354 temperature sections are summarized in Fig. 6(b) in the form of column. It can be found that  
 355 the average heat capacity flow rates of hot fluids in both the two sections are closer to those  
 356 of cold fluid as  $\dot{m}_{hh}$  decreases. **The more the number of intersection points is, the closer the**  
 357 **average heat capacity flow rates of the hot fluids are to that of cold fluid. The relationship**  
 358 **between the local heat capacity flow rates of hot and cold fluids is closely related to the local**  
 359 **performance, while the relationship between the average heat capacity flow rates is closely**  
 360 **related to the total performance.**

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363

364 Fig. 6. (a) The variation of local heat capacity flow rate with local heat transfer rate accumulation in  
 365 SHEs and (b) average heat capacity flow rate for cold and hot fluids at  $P_c = 7$  MPa and  $\tau = 65\%$ . **The**  
 366 **intersection points are marked with dots and numbers.**

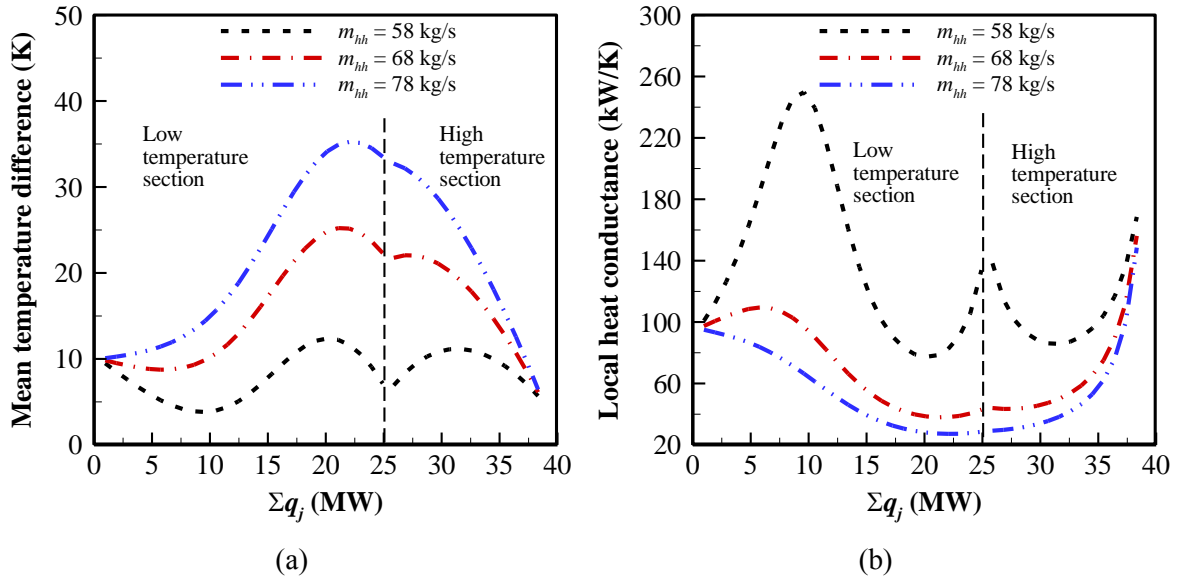
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368 The variations of local logarithmic mean temperature difference and local heat  
 369 conductance in the SHEs along the flow direction of cold fluid are elucidated in Fig. 7. The  
 370 local logarithmic mean temperature difference for each counter-flow SHE is defined as

$$\Delta T_m = \frac{(T_{hj,o} - T_{cj,i}) - (T_{hj,i} - T_{cj,o})}{\ln((T_{hj,o} - T_{cj,i}) / (T_{hj,i} - T_{cj,o}))} \quad (20)$$

371 Due to the fixed local heat transfer rate in each SHE, the local mean temperature difference  
 372 shows varying trends opposite to local heat conductance as shown in Fig. 7. The local mean  
 373 temperature difference and local heat conductance non-monotonically along the flow  
 374 direction of cold fluid in both the two temperature sections in the cases of smaller  $m_{hh}$ . The  
 375 position of the minimum local mean temperature difference or the required maximum local  
 376 heat conductance in the low temperature section moves toward the flow direction of cold  
 377 fluid as  $m_{hh}$  decreases. With the decrease in  $m_{hh}$ , the local mean temperature difference in  
 378 each SHE generally decreases, and accordingly the local heat conductance required by each  
 379 SHE increases in both the two temperature sections, whilst the heat conductance required by  
 380 the whole low temperature section exhibits larger increment than that required by the whole  
 381 high temperature section.

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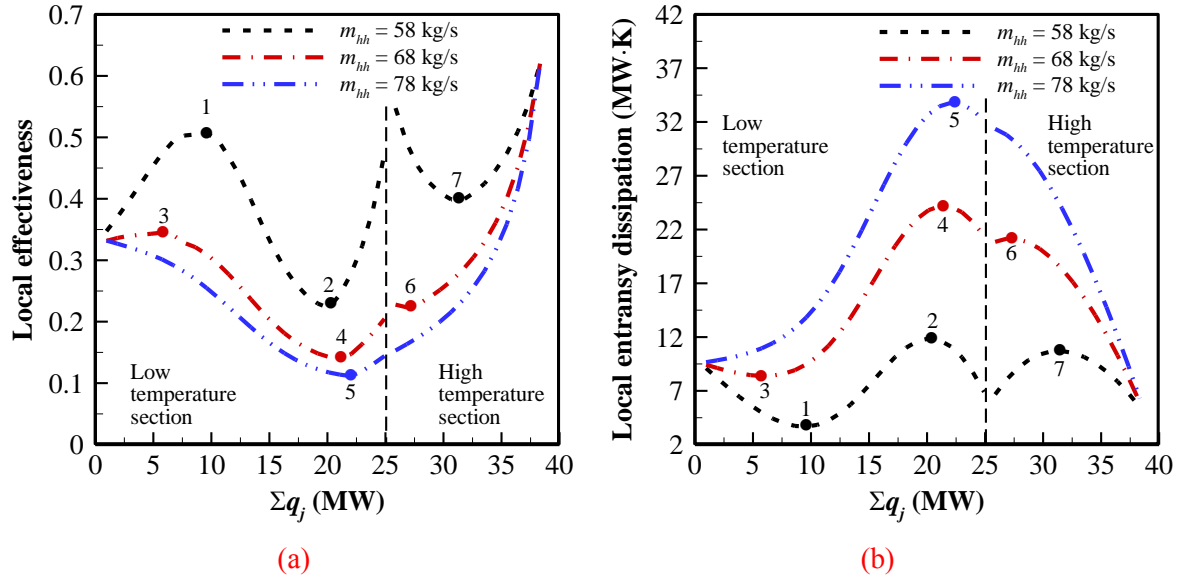
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385 Fig. 7. The variations of (a) local logarithmic mean temperature difference and (b) local heat  
 386 conductance with heat transfer rate accumulation in SHEs at  $P_c = 7$  MPa and  $\tau = 65\%$ .

387

388 Fig. 8 depicts the local effectiveness and local entransy dissipation in SHEs along the  
389 flow direction of cold fluid under different mass flow rates of hot fluids. According to the  
390 variations of the two local parameters, it can be found that the fall of  $m_{hh}$  generally improves  
391 the performance of each SHE under the same other conditions. The performance gap between  
392 different mass flow rates of hot fluids in the high temperature section decreases along the  
393 flow direction of cold fluid, while that in the low temperature section presents the varying  
394 trend similar to “N” shape as shown in Fig. 8(a). Comparing Figs. 8(b) and 7(a), we can find  
395 that the local entransy dissipation has very similar changing trend with the local mean  
396 temperature difference. This is because the irreversibility of heat transfer is primarily caused  
397 by the temperature difference when heat leak and flow friction loss is negligible. By  
398 comparing Fig. 8 with Fig. 6(a), the maximum local effectiveness and the minimum local  
399 entransy dissipation appear around the positions of  $R_{cj,hc} = 1$ , when the heat capacity flow  
400 rates of cold and hot fluids have the same changing tendency along the flow direction of cold  
401 fluids. On the contrary, if the two heat capacity flow rates have the opposite changing  
402 tendencies, around the positions of  $R_{cj,hc} = 1$  exist the minimum local effectiveness and the  
403 maximum local entransy dissipation. **Therefore, the same changing tendency of the heat**  
404 **capacity flow rates of two sides in the SHEs is beneficial for the improvement of local heat**  
405 **exchange performance.**

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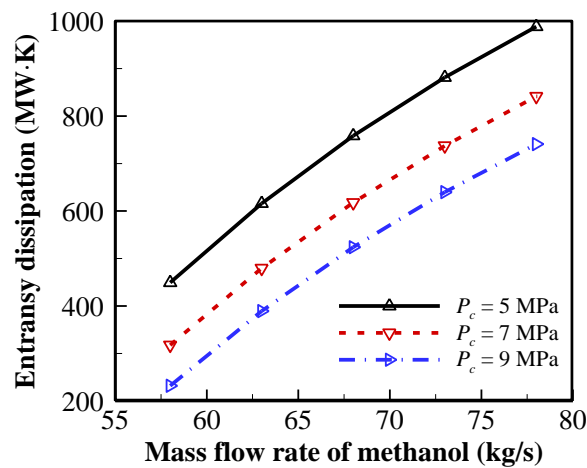
Fig. 8. The variations of (a) local effectiveness and (b) local entransy dissipation with heat transfer rate accumulation in SHEs at  $P_c = 7$  MPa and  $\tau = 65\%$ . The locations of extremums are marked with dots and numbers.

413 To evaluate the overall performance of T-N<sub>2</sub> evaporator, the variations of the total  
414 entransy dissipation and exergy efficiency with  $m_{hh}$  under different pressures of cold fluid  
415 are demonstrated in Figs. 9(a) and 9(b), respectively. The total entransy dissipation decreases  
416 as the pressure of cold fluid increases at the same mass flow rate and decreases with the  
417 decrease in  $m_{hh}$  at the same pressure, while the exergy efficiency shows totally opposite  
418 change trends. In combination with Fig. 6(b), it can be inferred that making the average heat  
419 capacity flow rate of hot fluids closer to that of cold fluid is beneficial to improving the heat  
420 exchange performance of the evaporator. By comparing Figs. 9(a, b) and 5(b), it is obvious  
421 that the smaller the temperature difference between hot and cold fluids at the junction of the  
422 two temperature section, the better the heat exchange performance of the evaporator. The  
423 calculation indicates that the extractable cold amount from the liquid N<sub>2</sub> with the same  
424 temperature rise from 83 K to 283 K decreases with the increase in the pressure of N<sub>2</sub>, which  
425 is caused by the decrease of its specific enthalpy as shown in Fig. 3(a). Specially, the  
426 extractable cold amount from unit mass of liquid N<sub>2</sub> decreases from 389.01 kJ/kg to 377.74



427 kJ/kg when the pressure increases from 5 MPa to 9 MPa. The manufacture cost of heat  
 428 exchanger is mainly determined by the heat conductance. Fig. 9(c) shows the variations of  
 429 required total heat conductance with  $m_{hh}$  under different pressures of cold fluid. It is clear  
 430 that the total heat conductance shows the varying tendencies opposite to the total entransy  
 431 dissipation number. This indicates that the improvement of the whole performance of  
 432 evaporator is at the expense of requiring larger area or/and coefficient of heat transfer. In  
 433 addition, the effect of the pressure of cold fluid on the required total heat conductance  
 434 gradually diminishes with the increase in  $m_{hh}$ .

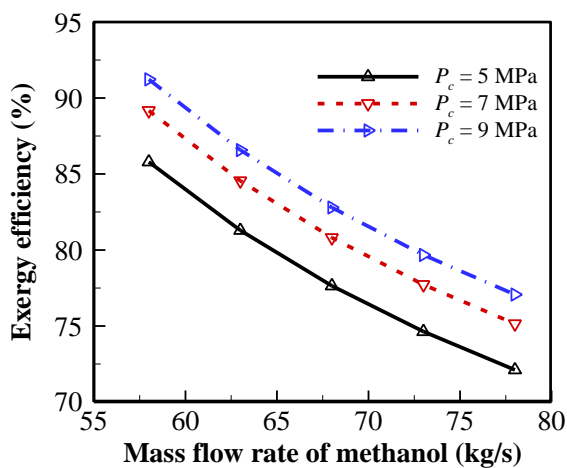
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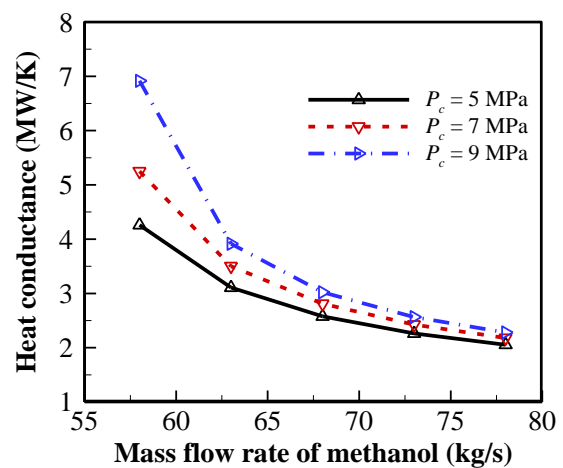
(a)



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(b)



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(c)

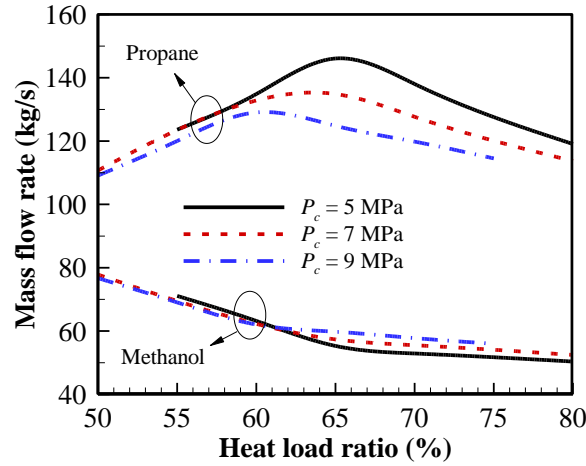
440 Fig. 9. The variations of (a) total entransy dissipation, (b) total exergy efficiency and (c) required total  
 441 heat conductance with mass flow rate of methanol under different pressures of cold fluid at  $\tau = 65\%$ .

442

443 In the above study, the heat load distribution ratio between the two temperature sections  
444 is selectively fixed. However, it is very pivotal for the performance of the evaporator and thus  
445 need to be elaborately considered in the design calculation. In order to compare heat  
446 exchange features under different heat load distribution ratios, the temperature difference  
447 between the hot and cold fluids at the junction of the two temperature section was kept 5 K. If  
448 this constraint condition cannot be satisfied at some heat load distribution ratios due to the  
449 assumption (g), the temperature of the hot fluids at the junction of the two-temperature  
450 section were fixed at 180.6 K, which is the lowest reliable operating temperature at the  
451 junction as mentioned in Section 2.3. Therefore, the minimum temperature difference  
452 between hot and cold fluids at the junction of the two temperature sections is achieved under  
453 the constraint of the assumption (g). The above conditions can be satisfied by adjusting the  
454 mass flow rates of propane and methanol. Based on the known data in Table 1, the mass flow  
455 rates of propane and methanol, the entransy dissipation and the exergy efficiency at different  
456 heat load ratios of the low temperature section among the whole evaporator are illustrated in  
457 Fig. 10. It can be found from Fig. 10(a) that the required mass flow rate of propane first  
458 increases and then decreases with the increase in the heat load ratio of the low temperature  
459 section, while that of methanol gradually decreases. Fig. 10(b) indicates that the minimum  
460 entransy dissipation can be achieved by adjusting heat load ratio of the low temperature  
461 section and it decreases with the increase in  $P_c$ . The heat load ratios of the low temperature  
462 section corresponding to the minimum values of entransy dissipation at  $P_c = 5$  MPa,  $P_c = 7$   
463 MPa and  $P_c = 9$  MPa are about 66%, 65% and 61%, respectively, which also correspond to  
464 the respective maximum mass flow rate of propane as shown in Fig. 10(a). By comparing  
465 Figs. 10(c) and 10 (b), it can be found that the exergy efficiency achieves the maximum as  
466 the entransy dissipation achieves the minimum under various pressures of cold fluids. It

467 manifests that the exergy efficiency and the entransy dissipation are congenerous in  
 468 performance evaluation of the evaporator when the heat load is fixed.

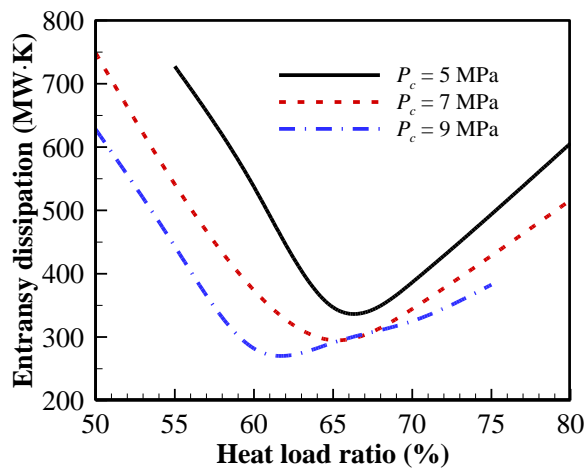
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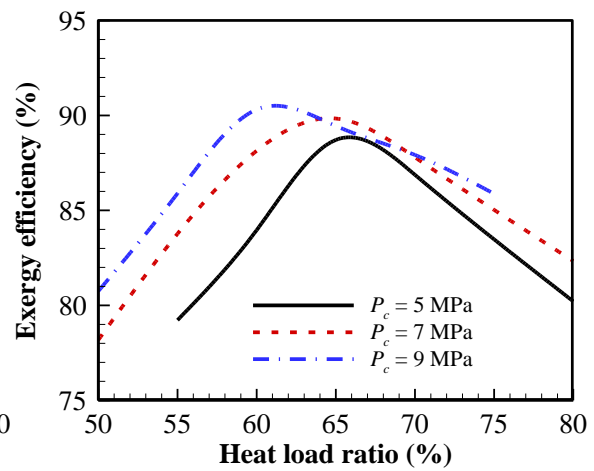
(a)



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(b)



(c)

474 Fig. 10. The effects of heat load ratio of low temperature section on (a) the required mass flow rates of  
 475 propane and methanol, (b) the entransy dissipation and (c) the exergy efficiency.

476

### 477 3.2. Check calculation

478 In this subsection, the total heat conductance of T-N<sub>2</sub> evaporator is given to analyze its  
 479 heat exchange performance and examine its affordable heat load. As listed in Table 1, the  
 480 total heat conductance is specified as  $4 \times 10^6$  W/K. The heat conductance ratio of the low  
 481 temperature section is set to 65% as an illustration. Similar to Fig. 5(a), the mass flow rate of

482 methanol is also determined by that of propane because of a temperature equality of two hot  
483 fluids at the junction of the two temperature sections in the assumption (f). The relations of  
484 the mass flow rates of the two hot fluids under different pressures of cold fluid are shown in  
485 Fig. 11(a). The required mass flow rate of methanol increases with that of propane under the  
486 given parameters of check calculation in Table 1. In order to ensure that the hot fluids stay  
487 liquid and avoid the invalidation of evaporator, the valid ranges of the mass flow rate of  
488 propane are different at different cold fluid pressures. For convenience of description, the  
489 mass flow rate of propane  $m_{hl}$  is used to indicate the variation of the mass flow rates of hot  
490 fluids in the following. Figs. 11(b) and 11(c) depict the temperature profiles of cold and hot  
491 fluids along the flow direction of cold fluid at different  $m_{hl}$ , respectively. The cold and hot  
492 fluids exhibit similar temperature profiles at the same  $m_{hl}$ . The temperature rise of cold fluid  
493 in the low temperature section diminishes with the increase in  $m_{hl}$ , which requires lower inlet  
494 temperature of propane; whereas the change is inverse in the high temperature section, which  
495 requires larger mass flow rate of methanol. Therefore, the eventual outlet temperature of cold  
496 fluid still increases with  $m_{hl}$ .

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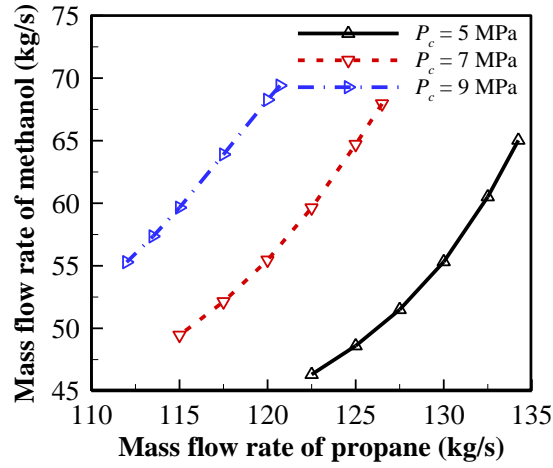
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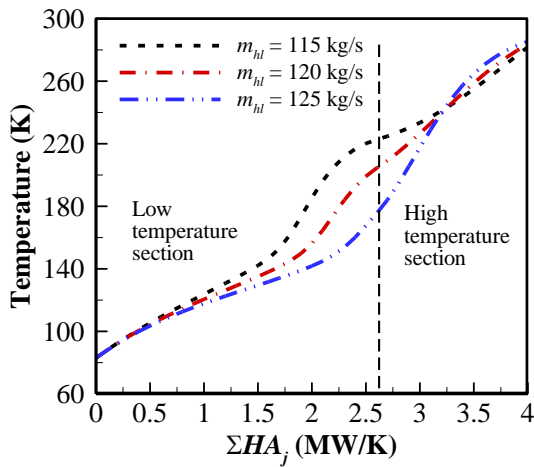
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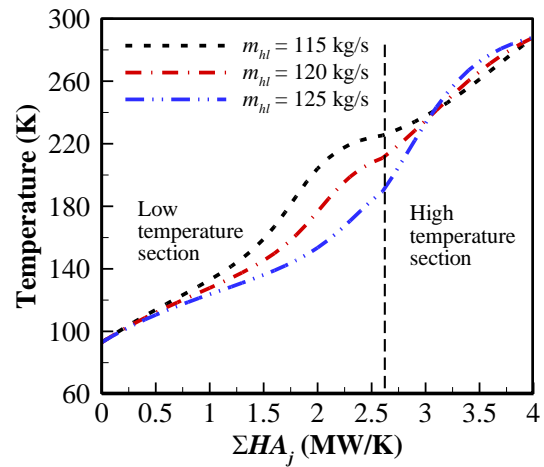
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(a)



(b)



(c)

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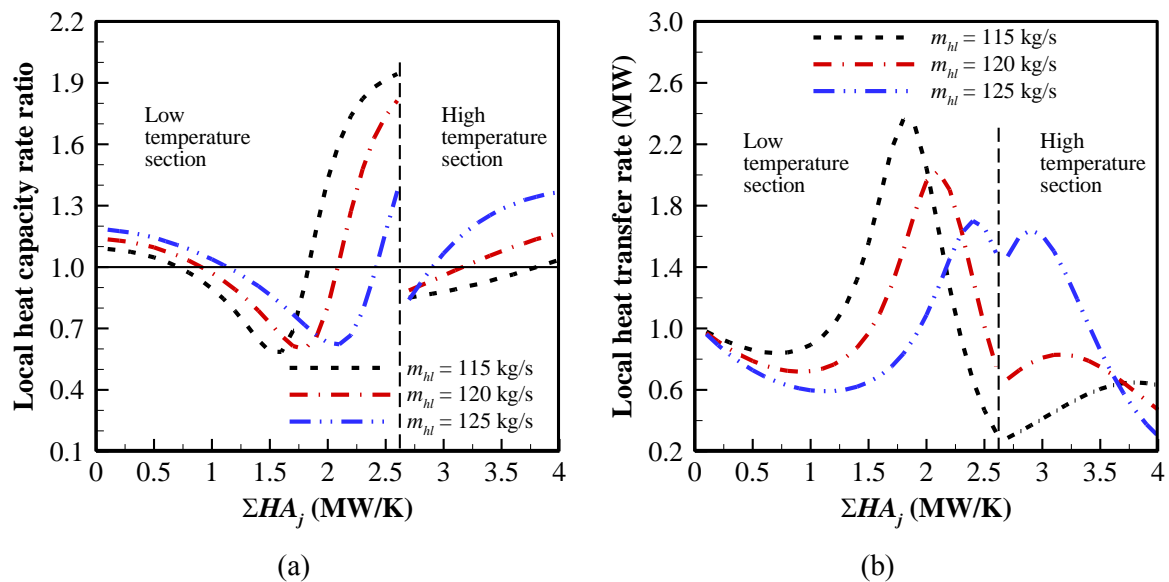
508 Fig. 11. (a) The relations of mass flow rates of the two hot fluids under different pressures of cold  
 509 fluid at  $\varphi = 65\%$ , and the temperature variations of (b) cold and (c) hot fluids with local heat  
 510 conductance accumulation in SHEs at  $P_c = 7$  MPa and  $\varphi = 65\%$ .

511

512 Fig. 12 elucidates the variations of local heat capacity rate ratio and local heat transfer  
 513 rate in the SHEs along the flow direction of cold fluid. Obviously, the local heat capacity rate  
 514 ratio first decreases and then increases along the flow direction of cold fluid in the low  
 515 temperature section, while it increases all along in the high temperature section as shown in  
 516 Fig. 12(a). The local heat capacity rate ratio also shows non-monotonic variation with the  
 517 increase in  $m_{hl}$ . This is because the fluids work at markedly different temperature regions at  
 518 different  $m_{hl}$  as shown in Fig. 11(b), which makes the fluids have notably different specific  
 519 heat as depicted in Fig. 1. As shown in Fig. 12(b), the gap of the local heat transfer rate

520 among different SHEs at the same  $m_{hl}$  increases with the decrease in  $m_{hl}$  in the low  
521 temperature section, while the situation is inverse in the high temperature section. The  
522 maximum gap at  $m_{hl} = 115$  kg/s is up to eightfold. Fig. 12(b) also indicates that there exist  
523 extremums of the local heat transfer rates in both the two sections, whose position and  
524 number are corresponding to those of  $R_{cj,hc} = 1$  as shown in Fig 12(a). The changing  
525 tendency of the local heat transfer rate exhibits reversals twice along the flow direction of  
526 cold fluid at different positions of  $R_{cj,hc} = 1$  in the low temperature section. Hence, the  
527 variation of local heat transfer rate highly dependent on the heat capacity rate ratio.

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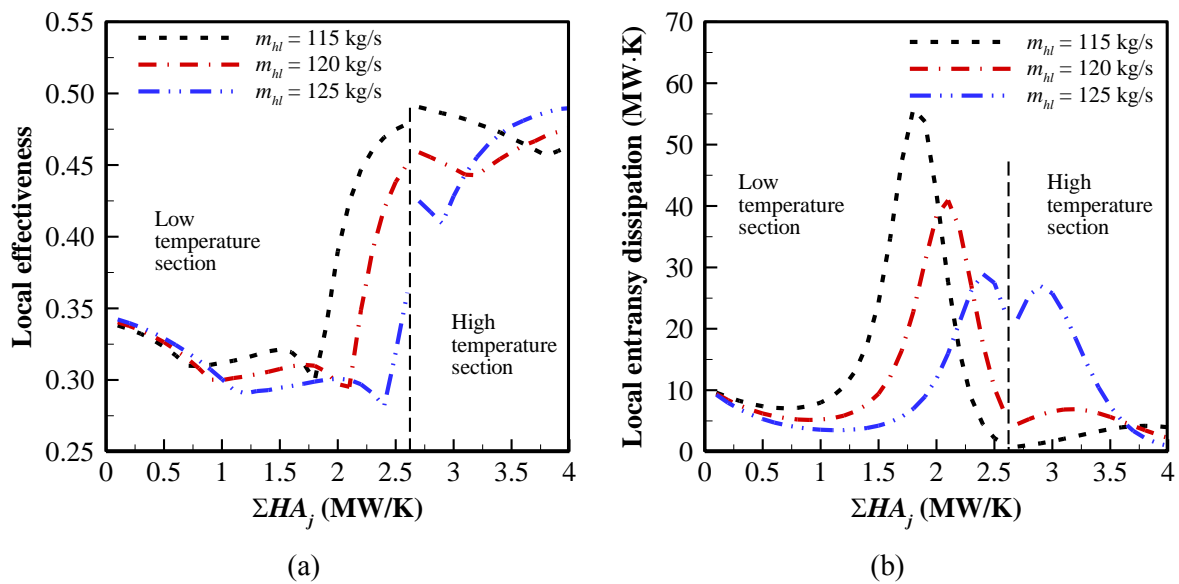
531 Fig. 12. The variations of (a) local heat capacity rate ratio and (b) local heat transfer rate with heat  
532 conductance accumulation in SHEs at  $P_c=7$  MPa and  $\varphi = 65\%$ .

533

534 The variations of the local effectiveness in the SHEs along the flow direction of cold  
535 fluid are illustrated in Fig. 13(a). In the low temperature section, the local effectiveness along  
536 the flow direction of cold fluid decreases before the first position of  $R_{cj,hc} = 1$  and sharply  
537 increases after the second position of  $R_{cj,hc} = 1$ , while it keeps nearly constant between the  
538 two positions; the local effectiveness tends to be constant with the increase in  $\dot{m}_{hl}$  before the

539 first position of  $R_{cj,hc} = 1$ , whilst it basically decreases after the first position of  $R_{cj,hc} = 1$ .  
540 In the high temperature section, the local effectiveness decreases before  $R_{cj,hc} = 1$  and  
541 increases after  $R_{cj,hc} = 1$ ; the local effectiveness has no explicit changing tendency with the  
542 increase in  $m_{hl}$ . Fig. 13(b) shows the variations of the local entransy dissipation with heat  
543 conductance accumulation in the SHEs. Along the flow direction of cold fluid, the local  
544 entransy dissipation shows changing trends basically similar to the local heat transfer as  
545 shown in Fig. 12(b). The larger the local heat transfer rate, the stronger the irreversibility of  
546 heat transfer. At smaller  $m_{hl}$ , the variation amplitude of local entransy dissipation in the low  
547 temperature section is notably larger than that in the high temperature section. As  $m_{hl}$   
548 increases, the maximum local entransy dissipation decreases in the low temperature section,  
549 while it increases in the high temperature section. This implies that the entransy dissipation  
550 shift from the low temperature section to the high temperature section as  $m_{hl}$  increases. The  
551 positions of maximum local entransy dissipation in the two temperature sections both move  
552 towards the junction of the two temperature sections with the increase in  $m_{hl}$ .

553



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556 Fig. 13. The variations of (a) local effectiveness and (b) local entransy dissipation with heat  
557 conductance accumulation in SHEs at  $P_c=7$  MPa and  $\varphi = 65\%$ .

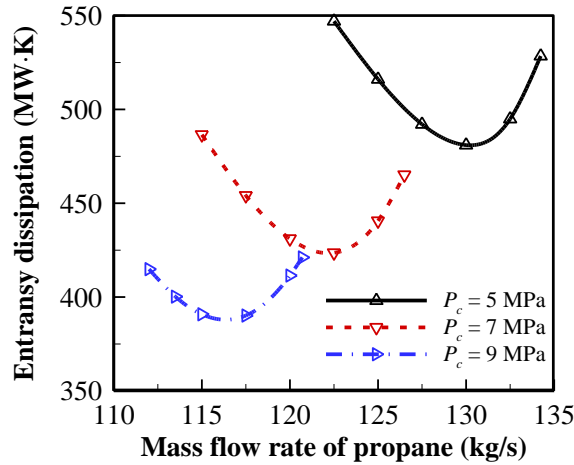
558 Fig. 14 displays the variations of total entransy dissipation, total exergy efficiency and  
559 affordable total heat load of the evaporator with  $m_{hl}$ . As shown in Figs. 14(a) and 14(b),  
560 there exist the minimum total entransy dissipation and the maximum total exergy efficiency  
561 at each given pressure of cold fluid and the corresponding  $m_{hl}$  decreases with the increase in  
562 the pressure of cold fluid. However, the  $m_{hl}$  corresponding to the maximum exergy  
563 efficiency is higher than that corresponding to the minimum entransy dissipation. Therefore,  
564 the exergy efficiency and the entransy dissipation cannot be considered the same indicator to  
565 the performance of the evaporator. As the pressure of cold fluid increases, the minimum  
566 entransy dissipation decreases and the maximum exergy efficiency increases. This manifests  
567 that a high cold fluid pressure is favourable for improving the heat exchange performance of  
568 evaporator and the optimum heat exchange performance can be achieved by adjusting the  
569 mass flow rates of the hot fluids. From Fig. 14(c), it can be seen that the affordable total heat  
570 load increases as  $m_{hl}$  increases in their respective valid ranges and decreases as the pressure  
571 of cold fluid increases. This implies that increasing the cold fluid pressure will leads to the  
572 diminution of the extractable cold amount from the liquid N<sub>2</sub> through the T-N<sub>2</sub> evaporator  
573 with a given total heat conductance.

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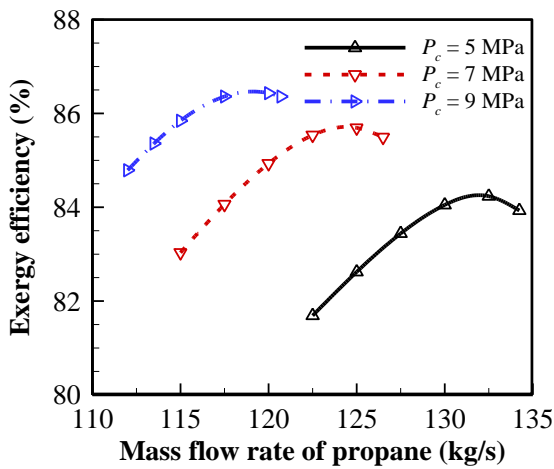
576





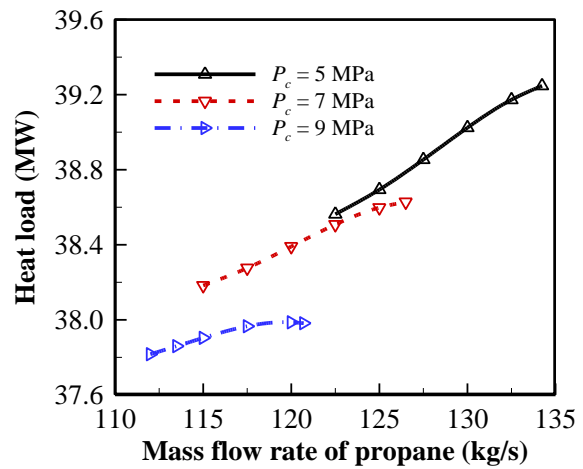
577  
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(a)



579  
580

(b)



579  
580

(c)

581 Fig. 14. The variations of (a) total entransy dissipation, (b) total exergy efficiency and (c) affordable  
582 total heat load with mass flow rate of propane under different pressures of cold fluid at  $\varphi = 65\%$ .

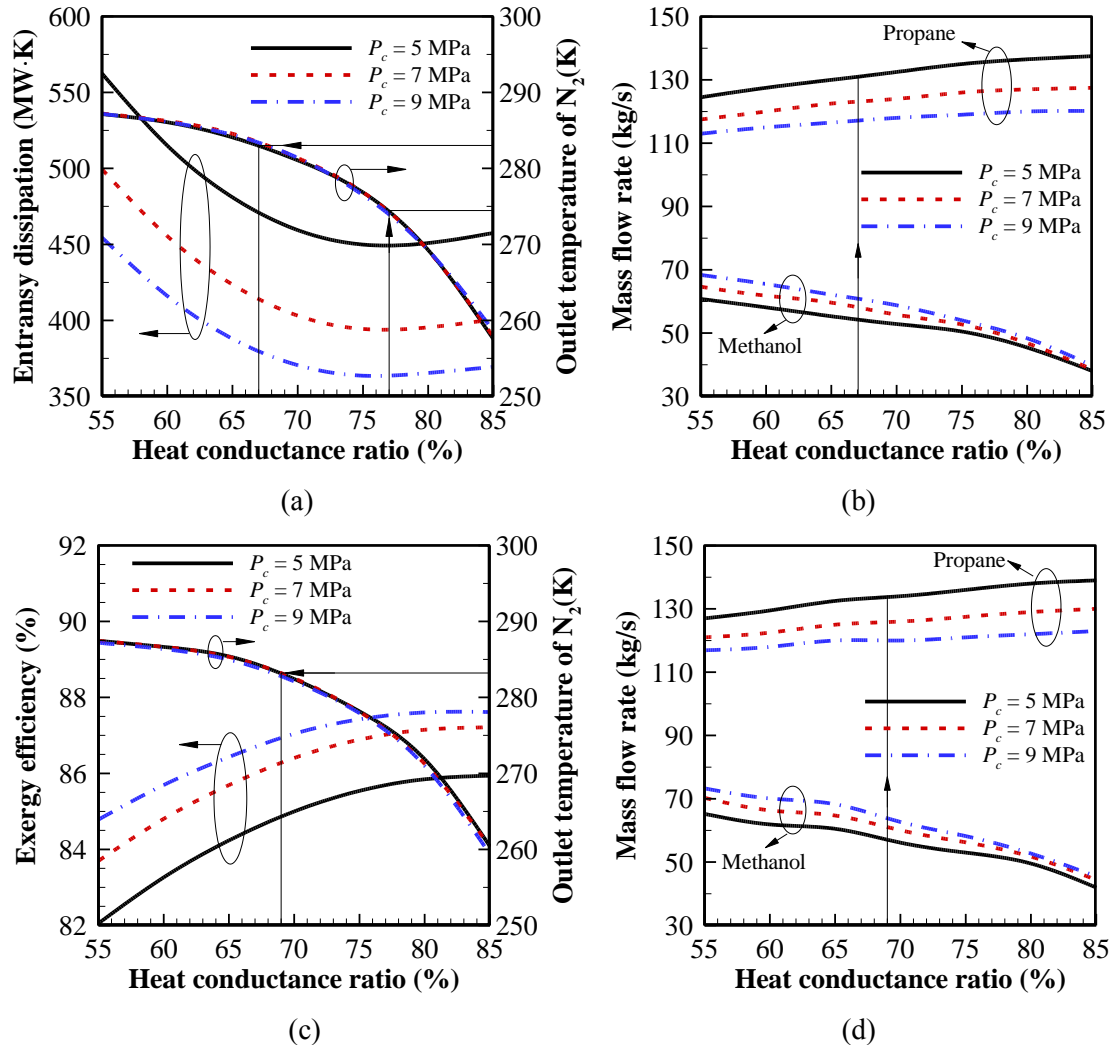
583

584 Similarly, the heat conductance distribution ratio between the two temperature sections  
585 is critical for the performance of the evaporator, which thus needs to be detailedly explored in  
586 the check calculation. The mass flow rates of the hot fluids were adjusted to obtain the  
587 respective minimum entransy dissipation and maximum exergy efficiency at various heat  
588 conductance distribution ratios for comparison. Fig. 15(a) depicts the variations of the  
589 minimum entransy dissipation and the corresponding outlet temperature of  $N_2$  with the heat  
590 conductance ratio of the low temperature section. Among the various ratios, the minimum  
591 entransy dissipation is the smallest at the ratio of about 77%, which slightly changes with the  
592 pressure of  $N_2$ . The outlet temperatures of  $N_2$  at the various pressures very close to each other

593 and decrease with the increase in the ratio. This means that heat exchange capacity  
594 diminishes with the increase in the ratio when keeping the respective minimum entransy  
595 dissipation. The outlet temperature of  $N_2$  is about 274 K at the ratio of 77%. Provided that the  
596 required outlet temperature is 283 K, the heat exchange capacity is insufficient when the ratio  
597 is set to 77%, although the entransy dissipation is the smallest under the circumstance.  
598 Simultaneously considering the demands for the outlet temperature of 283 K and smaller  
599 entransy dissipation, the heat conductance ratio of the low temperature section should be set  
600 as about 67% for various pressures of  $N_2$ . The required mass flow rates of propane and  
601 methanol corresponding to the minimum entransy dissipation at different heat conductance  
602 ratios of the low temperature section are illustrated in Fig. 15(b). It can be found that the  
603 required mass flow rate of propane increases with the increase in the heat conductance ratio  
604 of the low temperature section, while that of methanol gradually decreases. Fig. 15(c)  
605 summarized the maximum exergy efficiency and the corresponding outlet temperature of  $N_2$   
606 under various heat conductance ratios of the low temperature section. The maximum exergy  
607 efficiency progressively increases with the ratio and keeps nearly constant when the ratio is  
608 more than 80%, while the outlet temperatures of  $N_2$  decrease with the increase in the ratio.  
609 Simultaneously considering the demands for the outlet temperature of 283 K and larger  
610 exergy efficiency, the heat conductance ratio of the low temperature section should be set as  
611 about 69% for various pressures of  $N_2$ . The required mass flow rates of propane and methanol  
612 corresponding to the maximum exergy efficiency at various heat conductance ratios of the  
613 low temperature section are illustrated in Fig. 15(d). In comparison with Fig. 15(b), it can be  
614 seen that the required mass flow rates of propane and methanol to achieve the maximum  
615 exergy efficiency are both larger than those to achieve the minimum entransy dissipation. The  
616 above results indicate that the exergy efficiency and the entransy dissipation are not  
617 congenerous in performance evaluation of the evaporator when the heat conductance is fixed,

618 and therefore a compromise need to be made between the maximum exergy efficiency and  
 619 the minimum entransy dissipation.

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625 Fig. 15. The effects of heat conductance ratio of low temperature section: (a, b) minimum entransy  
 626 dissipation, corresponding outlet temperature of  $N_2$  and corresponding mass flow rates of propane  
 627 and methanol; (c, d) maximum exergy efficiency, corresponding outlet temperature of  $N_2$  and  
 628 corresponding mass flow rates of propane and methanol.

629

630

#### 631 4. Conclusions

632 The heat exchange performance analysis of T- $N_2$  evaporator used for cold storage or  
 633 recovery in the LAES system is implemented in this paper. Due to violent variation of

634 specific heat of  $N_2$  in the evaporator, the segmental design method is applied. The evaporator  
635 is divided into a low temperature section and a high temperature section according to two  
636 types of hot fluids (propane and methanol) used to receive the cold energy from the cold fluid  
637  $N_2$ .

638         When the total heat load is fixed, the local effectiveness and local heat conductance  
639 exhibit change trends opposite to local entransy dissipation along the flow direction of  $N_2$ .  
640 The local entransy dissipation achieves the minimum around the positions where the local  
641 heat capacity rate ratio equals one when the heat capacity flow rates of cold and hot fluids  
642 exhibit the same change trend along the flow direction of  $N_2$ , while it reaches the maximum  
643 around the positions when the two heat capacity flow rates exhibit opposite change trends.  
644 The total entransy dissipation decreases and the total exergy efficiency increases with the  
645 decrease in the mass flow rate of methanol or the increase in the pressure of  $N_2$ , while the  
646 required total heat conductance increases. The heat exchange performance of evaporator  
647 improves at the cost of heat conductance. The total entransy dissipation reaches the minimum  
648 and the total exergy efficiency achieves the maximum when about 66%, 65% and 61% of the  
649 total heat load is undertaken by the low temperature section at the  $N_2$  pressure of 5 MPa, 7  
650 MPa and 9 MPa, respectively. The extractable cold amount from the liquid  $N_2$  in the same  
651 temperature rise decreases with the increase in the  $N_2$  pressure.

652         When the total heat conductance is given, the required mass flow rate of methanol  
653 increases with that of propane. As the mass flow rate of propane increases, the required inlet  
654 temperature of propane decreases and the outlet temperature of  $N_2$  increases. The local heat  
655 capacity rate ratio equals one at several positions along the flow direction of  $N_2$ , where the  
656 change trends of local heat transfer rate, local effectiveness and local entransy dissipation are  
657 inverted. The minimum total entransy dissipation and the maximum total exergy efficiency  
658 can be achieved by adjusting the mass flow rate of propane, while their corresponding mass

659 flow rates of propane are different. Increasing the pressure of  $N_2$  is beneficial to lessening the  
660 minimum entransy dissipation and increasing the maximum exergy efficiency. The affordable  
661 heat load or cold amount from the liquid  $N_2$  increases as the mass flow rate of propane  
662 increases or the pressure of  $N_2$  decreases. The demands for the outlet temperature of 283 K  
663 and low entransy dissipation can be simultaneously satisfied when about 67% of the total heat  
664 conductance is distributed to the low temperature section for various pressures of  $N_2$ , while  
665 about 69% of the total heat conductance should be distributed to the low temperature section  
666 to achieve higher exergy efficiency.

667 The performance of T- $N_2$  evaporator cannot be intuitively predicted due to drastic  
668 variation of thermo-physical properties of  $N_2$  in the transcritical heat exchange process with  
669 wide working temperature range and multiple working pressure options. The operating  
670 parameters of T- $N_2$  evaporator should be carefully tailored to avoid its invalidation and  
671 elevate its overall performance.

672

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