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1 Experimental Study on a Multi-evaporator Refrigeration System with Variable Area  
2 Ratio Ejector

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7  
8 **Highlights:**

- 9 ● Experiments are conducted on a multi-evaporator refrigeration system (MERS).  
10 ● The energy saved by ejector is up to about 12% compared to conventional PRV.  
11 ● Sufficient superheat is conducive to high performance and low power consumption.  
12 ● Ejector with variable area ratio is an effective response to varying cooling load.

13 **Abstract:** This paper describes an experimental study on a multi-evaporator refrigeration system  
14 (MERS) with conventional pressure regulating valve (PRV) and variable area ratio ejector. Some key  
15 performance indicators such as cooling capacity, power consumption and ejector entrainment ratio  
16 were evaluated by switching operating modes, adjusting superheat and tuning ejector spindle and so on.  
17 The results indicated that: 1) the energy efficiency of the MERS can be improved by up to 12% by  
18 replacing the conventional pressure regulating valve with the variable area ratio ejector, 2) sufficient  
19 superheat or superheat degree greater than 2°C is conducive to high entrainment performance of the  
20 ejector, large cooling capacity of the low-temperature evaporator and low power consumption of the  
21 system, and 3) the effect of nozzle spindle position on the performance of the system is evident.

22 **Keywords:** Multi-evaporator refrigeration system; Variable area ratio ejector; Spindle; Cooling  
23 capacity allocation; Power consumption; Entrainment ratio

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## 25 **1. Introduction**

26 The term of *refrigeration* can be defined as a process that removes heat to cool  
27 articles or substances down to, and maintain them at a temperature lower than the  
28 ambient temperature. It has many applications including but not limited to: air  
29 conditioning, household refrigerators and industrial freezers. In the present society,  
30 refrigeration has become one of the most energy-intensive sectors. Estimates are  
31 difficult to give but as an average for the developed countries, its share in electricity  
32 use is thought to be between 10-20% [1]. For tropical countries such as Singapore,  
33 this ratio can even rise to over 50% due to the hot and humid weather [2].

34 According to the evaporating temperature, refrigeration applications could be  
35 divided into three categories: high-temperature refrigeration, medium-temperature  
36 refrigeration and low-temperature refrigeration. Air conditioning is a good example of  
37 high-temperature refrigeration. Fresh food storage is in the medium-temperature range,  
38 while storage of frozen and dairy products is typical application of low-temperature  
39 refrigeration which generally starts at 0 °C to as low as -30 °C [3]. In some  
40 circumstances, two or three evaporating temperatures can be involved. For example,  
41 in supermarket, approximately 50% of its total power consumption is associated with  
42 the medium and low temperature refrigeration [4-6], and 20% is used for air  
43 conditioning. Likewise, in applications such as food processing, transportation and  
44 storage, three evaporating temperatures, namely +7°C, - 5°C and - 30°C, are usually  
45 required for space cooling, storage of perishable or temperature sensitive food, and  
46 freezing, etc. [7-9]. A convenient way to provide such cooling demands is to design a

47 complete refrigeration unit for each temperature range, but this is not economical  
48 because of the high total investment. An alternative to cope with applications  
49 involving two or more evaporating temperatures is to use multi-evaporators  
50 refrigeration system (MERS), a single refrigeration system which consists of one  
51 compressor, one condenser but two or three evaporators with each working at  
52 different evaporating temperature[10].

53 The conventional MERS uses pressure regulating valve (PRV) to maintain the  
54 required pressure differences between the high-temperature and low-temperature  
55 evaporators. Considering the pressure losses caused by the throttle effect of PRV,  
56 efficiency improvements of multi-evaporators refrigeration system may be achieved  
57 by replacement of pressure regulating valve. Several schemes have been proposed, for  
58 example, multistage refrigeration cycles [11, 12], cascade refrigeration cycles [11, 12],  
59 expander based MERS [13-15] and ejector based MERS. The first three schemes are  
60 often used in ultra-low temperature refrigeration (cryogenic refrigeration) or CO<sub>2</sub>  
61 refrigeration systems where the pressure and temperature of the refrigerant drop  
62 greatly after expansion process. However, for applications of MERS involving fresh  
63 storage, freezing and air-conditioning, the energy saving by such schemes are not  
64 significant. The last scheme uses ejector in place of PRV to recover the pressure loss  
65 associated with throttling, and this seems to be an appropriate energy-saving  
66 alternative. For example, L. Kairouani, et al. [16] recently proposed an  
67 energy-efficient three-evaporator refrigeration system with two ejectors. The results of  
68 exergy analysis show that the system COP can be improved by about 10% for

69 two-temperature ejector-based MERS and 20% for three-temperature ejector-based  
70 MERS, respectively, as compared to the conventional MERS.

71 The main drawback of ejector based MERS is that it can only work well at the  
72 on-design condition. This is difficult to guarantee in actual operation since the cooling  
73 capacity allocation between evaporators depends not only on the cooling requirements  
74 of target application but also on the ambient climate variations. Ejector with variable  
75 area ratio may go far towards solving the problem [17]. Li et al. [18] conducted an  
76 experimental study on the MERS with a variable area ratio ejector , and concluded  
77 that ejector should have an area ratio smaller than the critical value to guarantee its  
78 pressure recovery effect. However, to the best knowledge of the authors, no  
79 information is available with respect to the effects of refrigerant physical state,  
80 namely saturated or superheated state, on the ejector performance as well as the  
81 MERS.

82 In the present work, the conventional PRV and a variable area ratio ejector are  
83 used and compared in order to better respond to the varying cooling load, and to  
84 achieve an efficient running of the MERS at a wider range of conditions. The main  
85 target of this study is to obtain the optimal refrigerant states and ejector geometry so  
86 that the MERS is capable of meeting different cooling requirements with low power  
87 consumption. These findings are expected to contribute to the energy savings of the  
88 MERS, and can be easily extended to applications such as supermarkets, retail stores,  
89 shipping containers, food processing and service equipment, and chemical industries.

## 90 **2. Experimental setup**

91 The multi-evaporator refrigeration system (MERS) developed in the present study  
92 is schematically shown in Fig. 1. The main components of this test facility are: a  
93 BITZER 4cc-9.2 semi-hermetic type of inverter compressor, an air-cooled condenser,  
94 two electronic expansion valves (indicated by EEV1 and EEV2, respectively), two  
95 testing chambers, a variable area ratio ejector, and a pressure regulating valve (PRV).  
96 The well accepted refrigerant R134a is used as the working fluid. All the equipment  
97 and pipes in the present system are well insulated by foam rubber to prevent heat  
98 losses.

99 The two testing chambers are used to simulate the fresh food storage room with  
100 evaporating temperature of  $-5^{\circ}\text{C}$  and the freezing room with evaporating temperature  
101 of  $-30^{\circ}\text{C}$ , respectively. Both chambers contain heat exchangers, blenders and power  
102 controllable heaters. The ethylene glycol solution is used for heat transfer with  
103 refrigerant, whose concentration is 20% (the corresponding freezing temperature is  
104  $-8^{\circ}\text{C}$ ) for chamber 1 and 50% (the corresponding freezing temperature is  $-37^{\circ}\text{C}$ ) for  
105 chamber 2. The blender is driven by electrical motor and helps to increase the heat  
106 transfer rate in the testing chamber. The power controllable heater is applied so that  
107 the cooling load in the testing chamber can be adjusted according to the cooling  
108 demand. When the temperature reaches stable state in testing chamber, the cooling  
109 capacity is just the same with the heater power.

110 Since the two chambers work at different evaporating temperatures and pressures,  
111 two approaches are adopted to maintain the required pressure difference between the  
112 high-temperature and low-temperature evaporators. In other words, the system can

113 run in two modes, i.e., the conventional mode and the pressure recovery mode. In the  
114 conventional mode, the PRV is used and the refrigerant vapor at higher temperature is  
115 first throttled and then compressed by the compressor. In the pressure recovery mode,  
116 a variable area ratio ejector characterized by its adjustable area ratio takes the place of  
117 PRV. Besides keeping the required pressure difference, and the incentive for  
118 application of variable area ratio ejector also includes: (a) to get required allocation of  
119 cooling capacity by adjusting the entrainment rate of secondary fluid, and (b) to  
120 improve the system energy efficiency by the pressure recovery effect of the ejector  
121 and by the higher compressor efficiency resulting from the reduced compression ratio.

122 All the pressures are measured by pressure transducers with the accuracies of 0.5%  
123 of full scales. The temperatures are measured by PT1000 platinum resistance with an  
124 error of  $\pm 0.3^{\circ}\text{C}$ . The flow rates are measured by two metal tube rotameters: one is  
125 mounted before EEV1 for refrigerant passing through the high-temperature  
126 evaporator, and the other is located before EEV2 for low-temperature evaporator.  
127 Each rotameter has an accuracy of  $\pm 1.6\%$ .

128 The output signals from the measurement devices are transferred to a PC through  
129 a data acquisition board. The automatic monitor, control and data acquisition system  
130 is developed in Labview 2010 of the National Instruments [11], and can display the  
131 real-time working parameters and make record on demand. For a view of the real  
132 experimental rig and the monitor panel, see Fig. 2.

### 133 **3. Geometric parameters of the variable area ratio ejector**

134 An illustration of the variable area ratio ejector has been presented in Fig. 3. As

135 shown, it consists of five parts: the primary nozzle, the motor-driven spindle, the  
 136 suction chamber, the mixing chamber, and the diffuser. The main geometry  
 137 parameters of the ejector are designed according to refs. [19-21]. The  
 138 constant-pressure section of mixing chamber has a converging angle of  $25^\circ$ , while the  
 139 constant-area section has a length ( $L$ ) of 40 mm and a diameter ( $D$ ) of 10mm. The  
 140 primary nozzle used in the present study has an inlet diameter ( $d_1$ ) of 15mm, throat  
 141 diameter ( $d_2$ ) of 4mm and exit diameter ( $d_3$ ) of 6mm. It is placed upstream of the  
 142 mixing chamber, and the distance between the exit of primary nozzle and the start of  
 143 constant-area mixing chamber is 15mm. The motor-driven spindle is used for  
 144 adjustment of primary nozzle. When the spindle moves, the effective flow area of  
 145 nozzle throat changes, so does the ejector area ratio. The movement of spindle is  
 146 described by the spindle position  $x$ , which is defined as the relative position of  
 147 spindle tip in the primary nozzle. As illustrated in Fig. 3,  $x$  is negative when the  
 148 spindle tip is in the convergent part of primary nozzle, equal to zero when the spindle  
 149 tip moves to the throat section, and positive when the spindle tip is in the divergent  
 150 part. And the relation between the spindle position  $x$  and the area ratio  $AR$  is as  
 151 follows:

$$152 \quad AR = \begin{cases} 6.25 & , \quad x \leq 0 \\ \frac{5625}{900 - 4x^2} & , \quad 0 < x < 15 \end{cases} \quad (1)$$

153 The refrigerant coming from the high-temperature evaporator (Evaporator 1)  
 154 enters the primary nozzle as the high-pressure primary fluid, while the refrigerant  
 155 from the low-temperature evaporator (Evaporator 1) works as the entrained secondary



156 fluid. By moving the spindle inside the primary nozzle, it is possible to regulate the  
157 flow rates of primary and secondary fluids, which is an efficient adjustment in  
158 response to the variable cooling load .

#### 159 **4. Effect of working modes on the performance of MERS**

160 As mentioned in Section 2, the multi-evaporator refrigeration system can run in  
161 two working modes, i.e., the conventional mode and the ejector-based pressure  
162 recovery mode. The two modes differ in the way of maintaining the required pressure  
163 difference between the high-temperature and low-temperature evaporators. The  
164 former uses a PRV to reduce the pressure of the high-pressure refrigerant stream  
165 before it mixes with the low-pressure stream, while the latter uses a variable area ratio  
166 ejector to accomplish both the decompression and mixing processes.

167 Since the two evaporators are used for different applications (Evaporator 1 for  
168 refrigeration and Evaporator 2 for freezing, respectively) and operate at different  
169 pressure and temperature levels, the power consumption rather than the coefficient of  
170 performance (COP) is used to evaluate the system performance. Specifically, the  
171 power consumed by the system is compared when each evaporator provides equal  
172 cooling capacity in both working modes.

173 A comparison experiment is carried out between the conventional mode and the  
174 pressure recovery mode. The results are shown in Fig. 4. It is found that power  
175 consumption increases linearly with the cooling capacity of Evaporator 1 in both  
176 conventional and pressure recovery modes. And the power consumption of  
177 conventional mode is higher than that of the ejector-based pressure recovery mode.

178 This means that the utilization of ejector helps to improve the energy saving effect of  
179 the whole system. The main reason is that refrigerant vapor at higher temperature  
180 suffers a pressure drop before it is compressed by the compressor, and compared with  
181 throttling of pressure regulating valve, ejector can recover part of the pressure loss  
182 associated with evaporating pressure difference. As a result, the compression ratio of  
183 compressor decreases while the compressor efficiency increases, which are beneficial  
184 to the reduction of power consumption.

185 It can also be concluded from Fig. 4 that the higher the cooling capacity of  
186 Evaporator 1 is, the more energy saving can be achieved by the ejector-based pressure  
187 recovery mode. For example, when the cooling capacity of Evaporator 1 is 1.0kW, the  
188 power consumption of pressure recovery mode is 1.95kW, saving about 2% in  
189 comparison with 1.99kW of the conventional mode. As the cooling capacity of  
190 Evaporator 1 increases to about 2.3kW, the power saved by the pressure recovery  
191 effect of ejector increases to about 12%. In other words, larger flow rate of  
192 high-temperature refrigerant stream makes the system more energy efficient. This is  
193 mainly because the pressure loss recovered by ejector is part of the pressure drop of  
194 the high-pressure refrigerant, and it increases with the increasing flow rate of  
195 high-pressure refrigerant since the pressure drop keeps constant at fixed evaporator  
196 pressures.

## 197 **5. Effect of ejector primary fluid state on the performance of MERS**

198 It is an important feature of the R134a T-S coordinate diagram that its specific  
199 entropy decreases with the rising temperature along the saturated vapor line. This

200 means that isentropic expansion of R134a from saturated vapor line will enter the  
201 wet-vapor region. As for ejector with R134a as working fluid, condensation may  
202 occur if the primary fluid is in or near the saturation state. This phenomenon could  
203 make quite an impact on the performances of ejector and MERS. In this section,  
204 superheat degree is considered as an indicator of primary flow physical state, and the  
205 dependencies of both system and ejector performances on superheat heat degree are  
206 analyzed in order to get a better understanding.

#### 207 (1) System performance

208 The cooling capacity varies with the physical state of primary fluid. Take the  
209 operation condition listed in Table 1 as an example. The physical state of ejector  
210 primary flow changes from saturated state to superheated state when the ethylene  
211 glycol solution in the high-temperature evaporator is subjected to a sustainable  
212 temperature rise. In this process, the cooling capacities of both evaporators are  
213 strongly influenced. Fig. 5(a) shows a typical view of the cooling capacity variations  
214 with primary flow superheat degree, and the ejector geometry involved has a spindle  
215 position of -5mm and an area ratio of 6.25. The first and most obvious feature of these  
216 two cooling capacity curves is that the cooling capacity keeps almost constant as the  
217 superheat degree varies except close to the saturated point where drastic variation is  
218 observed. To be specific, the cooling capacity of the high-temperature evaporator  
219 (Evaporator 1) drops from 2.4kW to 1.9kW when the superheat degree increases from  
220 0 to 2°C, and then keeps around 2.0 kW when superheat degree is further increased.  
221 Meanwhile, the cooling capacity of the low-temperature evaporator (Evaporator 2)

222 increases sharply from 0.7kW and then approaches a constant value of 1.1kW. Similar  
223 variations are observed for  $x=0, 5\text{mm}, 10\text{mm}$  ( $AR=6.25, 7.03, 11.25$ , respectively).

224 The power consumption of compressor varies in a way similar to the cooling  
225 capacity of Evaporator 1. That is, it first decreases rapidly and then keeps almost  
226 constant. This is in accordance with the linear relation of power consumption obtained  
227 in Section 3, and makes it possible to conclude that superheat is beneficial to energy  
228 saving.

## 229 (2) Ejector performance

230 The ejector performance is evaluated by the entrainment ratio which is defined  
231 as:

$$232 \quad ER = \frac{m_s}{m_p} \quad (2)$$

233 where  $m_p$  is the mass flow rate of primary fluid, and  $m_s$  is the mass flow rate of  
234 secondary fluid. The entrainment ratio reflects to some extent the cooling capacity  
235 allocation between the two evaporators. The higher the entrainment ratio, the larger  
236 cooling capacity the low-temperature evaporator provides.

237 As shown in Fig 5(b), the entrainment ratio is strongly influenced by the  
238 superheat degree of primary flow. The entrainment ratio rises remarkably (from 0.25  
239 to 0.55) when the superheat degree increases from 0 to 2°C but little difference is  
240 observed when superheat degree is further increased. Similar variation of entrainment  
241 ratio to superheat degree are also founded for  $x=0, 5, 10$  (see Fig 5(c)). It should be  
242 noted that the rapid increase of entrainment ratio is caused by the decreasing  $m_p$  and  
243 the increasing  $m_s$ . Therefore superheat is required to ensure high entrainment

244 performance and accordingly large cooling capacity of low-temperature evaporator. In  
245 other words, it is favorable to keep the primary flow under superheat condition from  
246 the viewpoint of exergy and energy quality.

247       Considering the different behavior of MERS and ejector, it is convenient to divide  
248 the primary flow states into two categories: the saturated state and the superheated  
249 state. The former one refers to the primary flow states close to the saturation curve  
250 and has a superheat degree less than  $2^{\circ}\text{C}$ . And the latter refers to the primary flow  
251 states with a certain degree of superheat, namely, the dry-vapor state. The temperature  
252 range of saturated state is a bit narrow comparing with the total error (about  $0.6^{\circ}\text{C}$ ),  
253 which is calculated based on error propagation by taking into account not only the  
254 error of temperature sensor ( $0.3^{\circ}\text{C}$ ) but also the contribution of pressure sensor to  
255 error of saturated temperature (about  $0.55^{\circ}\text{C}$  at  $0.24\text{MPa}$  for pressure sensor with a  
256 range of  $0\sim 1.0\text{MPa}$  and an accuracy of  $0.5\%$ ). However, the differences of system and  
257 ejector performances are fairly obvious between superheated and saturated conditions,  
258 and can be observed for different ejector configurations. At saturated primary state,  
259 variation of superheat degree causes rapid changes of cooling capacity, power  
260 consumption and entrainment ratio. While at superheated primary state, the effect of  
261 superheat degree is relatively small. This may be resulting from the fact whether the  
262 condensation process takes place in the primary nozzle. Since the saturated vapor line  
263 of R134a has a negative slope characteristic, the saturated primary fluid will enter the  
264 wet-vapor region when it expands inside the nozzle, and erosive droplets of high  
265 velocity appear accordingly. And the sensitivity of system performance to superheat

266 degree is considered to be a direct response to the generation of liquid droplets. On  
267 the other hand, if the inlet vapor of primary nozzle is superheated enough, the fluid  
268 remains in the dry-vapor state after expansion. In this case, the system performance  
269 appears to be independent of superheat degree.

## 270 **6. Effect of ejector spindle position on the performance of MERS**

271 As mentioned in section 3, the spindle in primary nozzle is movable under the  
272 driving of motor. When the spindle moves forward over the position  $x = 0$ , the  
273 effective flow area of nozzle throat will decrease, which leads to a rapid increase of  
274 area ratio.

275 The variation of ejector entrainment ratio with spindle position  $x$  is given in Fig. 6  
276 and the corresponding test conditions are shown in Table 2. It can be seen that when  
277 the spindle position  $x$  varies between -5~0mm ( $AR$  maintains itself at a constant  
278 value of 6.25), entrainment ratios under both saturated and superheated conditions are  
279 almost kept unchanged. As the spindle moves forward further to 7.5 (the  
280 corresponding  $AR$  is 8.33), the entrainment ratio under superheated condition  
281 increases from 0.6 to 1.6 while the entrainment ratio under saturated condition  
282 increases from 0 to 0.75. This suggests that it is the area ratio of ejector rather than the  
283 spindle position that basically determines the variation of entrainment ratio.  
284 Furthermore, the entrainment ratio under superheated conditions is higher than that  
285 under saturated condition. In superheated conditions, therefore, the cooling capacity  
286 of low-temperature evaporator (Evaporator 2) makes up a larger portion when  
287 compared with the saturated condition. This is consistent with the results presented in

288 Section 5.

289 The effect of nozzle spindle position on the cooling capacity is depicted in Fig. 7  
290 with (a) for high-temperature evaporator (Evaporator 1) and (b) for low-temperature  
291 evaporator (Evaporator 2). Still the cooling capacity keeps constant as the spindle  
292 moves in the primary nozzle until the area ratio of ejector begins to change. The  
293 cooling capacity of high-temperature evaporator (Evaporator 1) has a tendency  
294 qualitatively opposite to the entrainment ratio curve. It first keeps constant, say,  
295 2.5kW for the saturated case, and then decreases with the spindle position. When the  
296 spindle tip moves to the position of 7.5mm (the corresponding  $AR$  is 8.33), the  
297 cooling capacity of high-temperature evaporator under saturated condition goes down  
298 to around 1.6kW. For the case of superheated primary state, this cooling capacity  
299 decreases gradually from 2kW to 1.25kW when the spindle position increases from a  
300 value of 0mm to 7.5mm. Unlike the high-temperature evaporator, the low-temperature  
301 evaporator (Evaporator 2) provides a cooling capacity proportional to the area ratio.  
302 When the spindle moves from 0mm to 7.5mm with the area ratio increasing from 6.25  
303 to 8.33, the cooling capacity of Evaporator 2 increases from 0 to 1.25kW in saturated  
304 case and to 1.625kW in superheated case. Moreover, the cooling capacity of  
305 high-temperature evaporator under superheat condition is lower than that under  
306 saturated condition while the cooling capacity of low-temperature evaporator presents  
307 an opposite behavior.

308 Therefore, from a viewpoint of energy saving, large area ratio is recommended  
309 for high low-temperature cooling and low power consumption.

## 310 **7. Conclusions**

311 This paper is concerned with a multi-evaporator refrigeration system (MERS)  
312 that uses R134a as working fluid. The two evaporators of MERS are used for  
313 refrigeration and freezing, respectively. Efforts have been targeted at the feasibility of  
314 replacing the conventional PRV with a variable area ratio ejector to regulate their  
315 cooling capacity allocation and to recover the pressure loss. The primary findings can  
316 be summarized as follows:

317 (1) Energy efficiency improvements of MERS are achieved by replacing the  
318 conventional PRV with an ejector. In the range of cooling capacity measured  
319 in the present study, the energy saved by the pressure recovery effect of ejector  
320 is up to about 12%.

321 (2) The power consumption increases linearly with the cooling capacity of  
322 high-temperature evaporator (Evaporator 1) for both conventional and  
323 ejector-based pressure recovery working modes.

324 (3) Sufficient superheat or superheat degree greater than  $2^{\circ}\text{C}$  is required to ensure  
325 high entrainment performance, large cooling capacity of low-temperature  
326 evaporator (Evaporator 2) and low power consumption.

327 (4) The cooling capacity of the system shows a strong dependence on the area  
328 ratio of ejector. When the spindle moves from 0mm to 7.5mm with the area  
329 ratio increasing from 6.25 to 8.33, the cooling capacity of Evaporator 2  
330 increases from 0 to 1.25kW in saturated case and to 1.625kW in superheated  
331 case.



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336 **Nomenclatures**

- 337  $d_1$  inlet diameter of primary nozzle, mm  
338  $d_2$  throat diameter of primary nozzle, mm  
339  $d_3$  outlet diameter of primary nozzle, mm  
340  $D$  diameter of constant-area mixing chamber, mm  
341  $L$  length of constant-area mixing chamber, mm  
342  $m_p$  mass flow rate of primary flow,  $\text{kg}\cdot\text{s}^{-1}$   
343  $m_s$  mass flow rate of secondary flow,  $\text{kg}\cdot\text{s}^{-1}$   
344  $p_p$  primary flow pressure of ejector, bar  
345  $p_s$  secondary flow pressure of ejector, bar  
346  $p_b$  outlet pressure of ejector, bar  
347  $x$  the relative spindle position, mm

348 **Abbreviation**

- 349  $AR$  ejector area ratio,  $D^2\cdot d_2^{-2}$   
350  $ER$  entrainment ratio,  $m_s\cdot m_p^{-1}$   
351 MERS multi-evaporator refrigeration system  
352 PRV pressure regulating valve

353 **Index**

354 N1 inlet section of primary nozzle  
355 N2 throat section of primary nozzle  
356 N3 outlet section of primary nozzle

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429 **Figure captions**

430 Fig. 1 Schematic view of the experimental rig

431 Fig. 2 Photograph of the experimental rig and user interface: (a) Experimental rig,

432 (b) Monitor and control panel

433 Fig. 3 Schematic diagram and dimension of the variable area ratio ejector and

434 primary nozzle

435 Fig. 4 Comparison of conventional working mode with pressure recovery mode

436 Fig. 5 Entrainment ratio and cooling capacity versus superheat degree: (a) cooling

437 capacity ( $x = -5$ ), (b) entrainment ratio ( $x = -5$ ) and (c) entrainment ratio

438 ( $x = 0, 5, 10$ )

439 Fig. 6 Variation of ejector entrainment ratio with spindle position at saturated and

440 superheated conditions

441 Fig. 7 Variations of cooling capacity with spindle position at saturated and

442 superheated conditions (a) cooling capacity of Evaporator 1, (b) cooling

443 capacity of Evaporator 2

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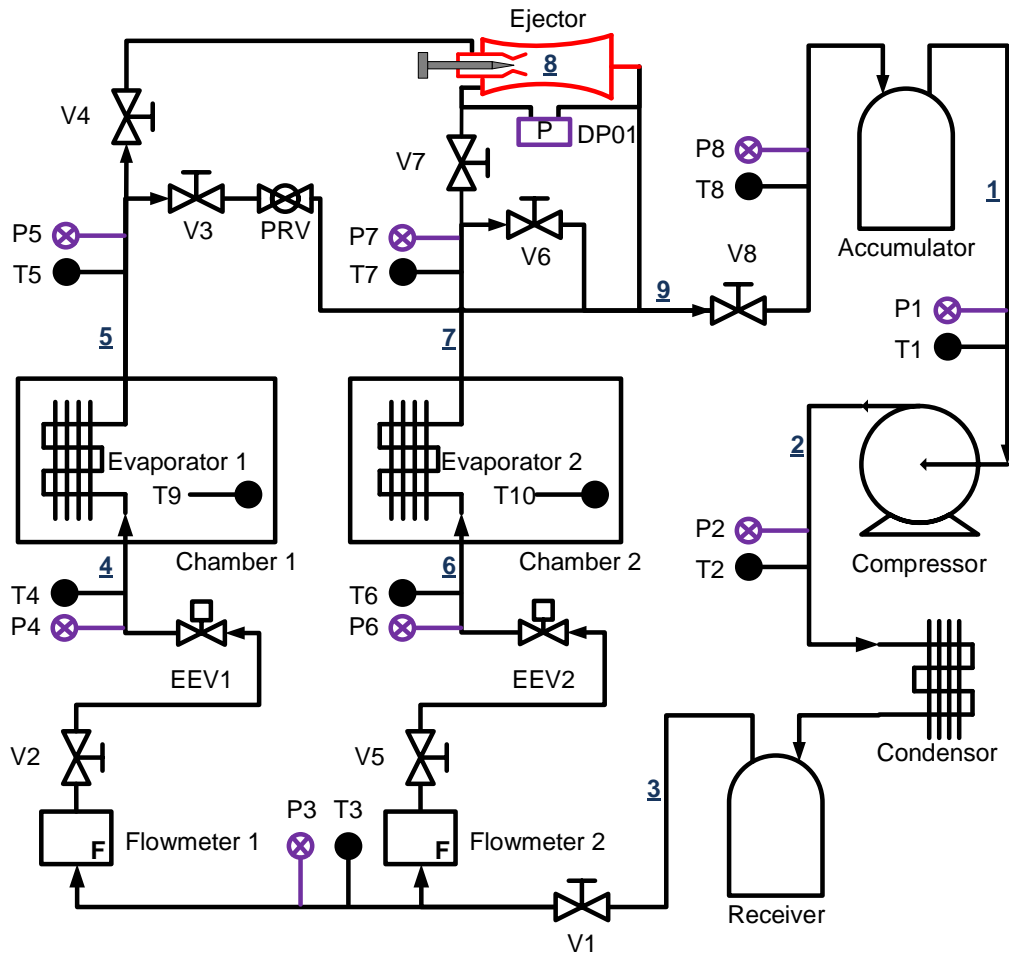
446 **Figures**

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452 Fig. 1 Schematic view of the experimental rig

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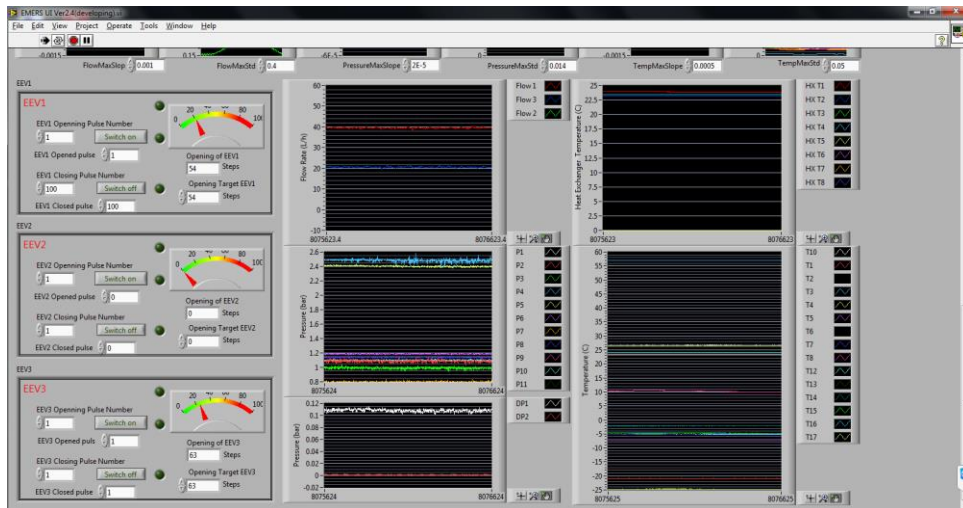
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(a) Experimental rig



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(b) Monitor and control panel

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Fig. 2 Photograph of the experimental rig and user interface

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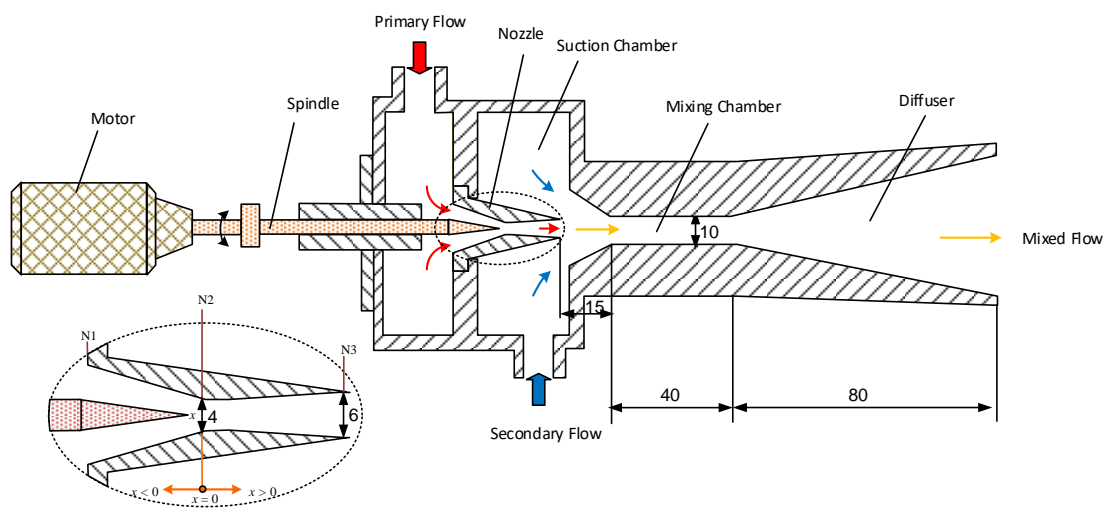
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Fig. 3 Schematic diagram and dimension of the variable area ratio ejector

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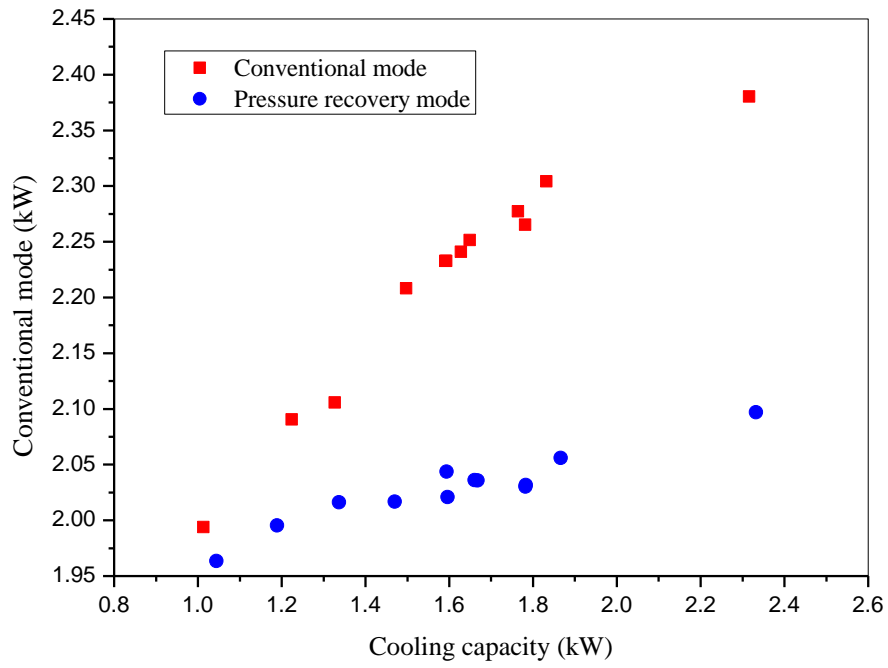
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491 Fig. 4 Comparison of conventional working mode with pressure recovery mode

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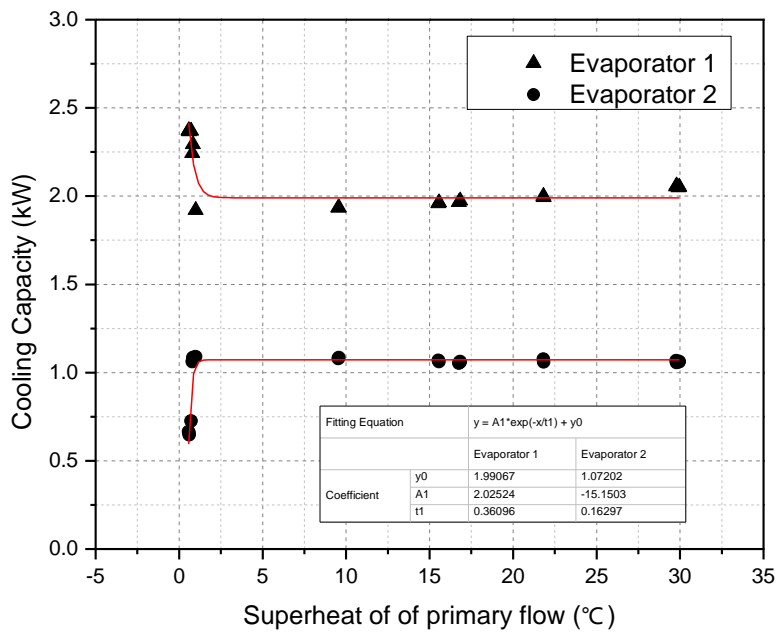
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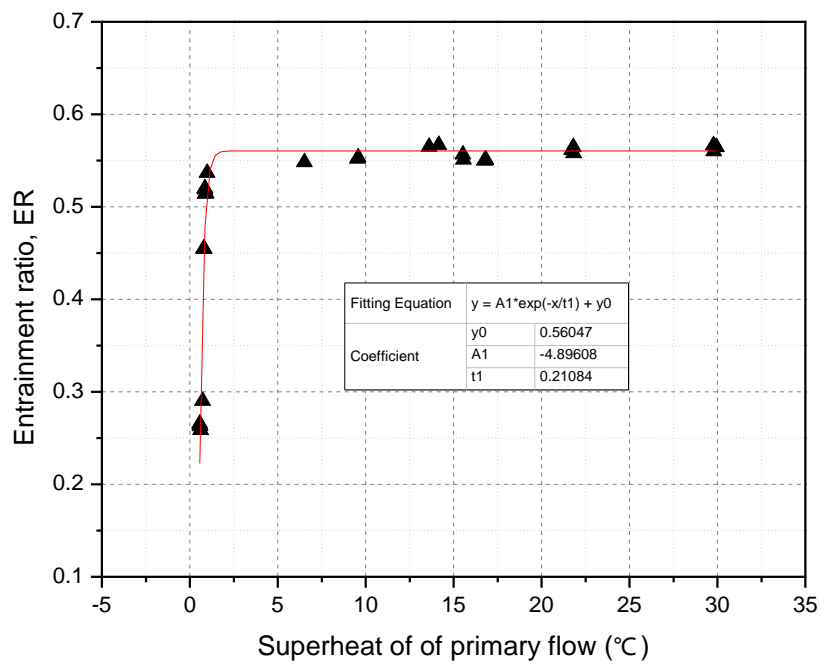




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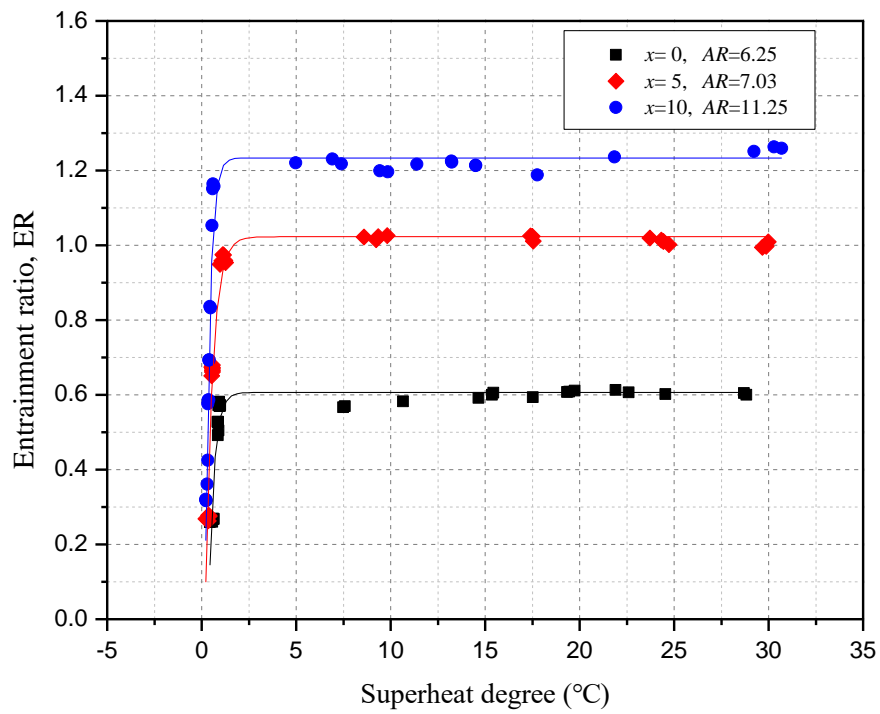
(a)



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(b)



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(c)

503 Fig. 5 Entrainment ratio and cooling capacity versus superheat degree: (a) cooling capacity

504 ( $x = -5$ ), (b) entrainment ratio ( $x = -5$ ) and (c) entrainment ratio ( $x = 0, 5, 10$ )

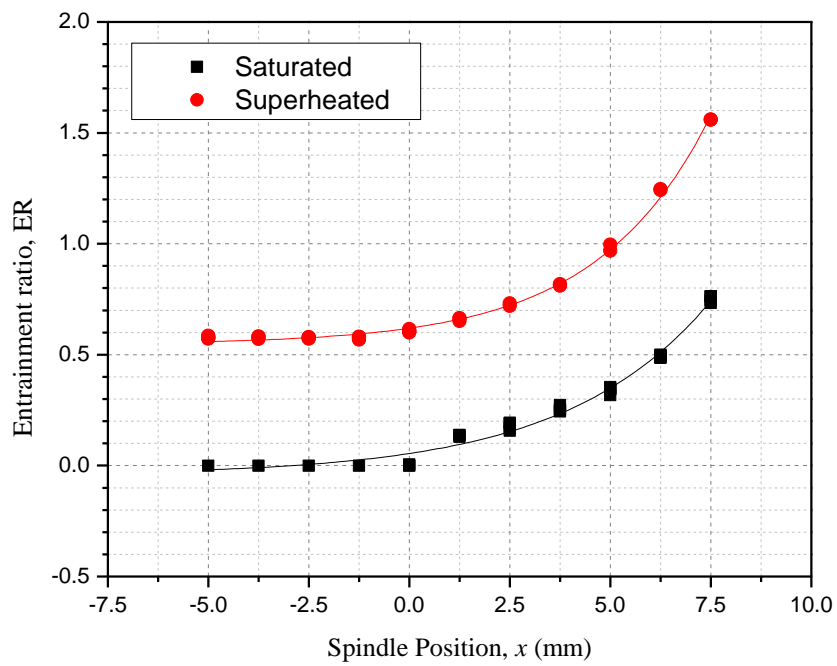
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512 Fig. 6 Variation of ejector entrainment ratio with spindle position at saturated and superheated

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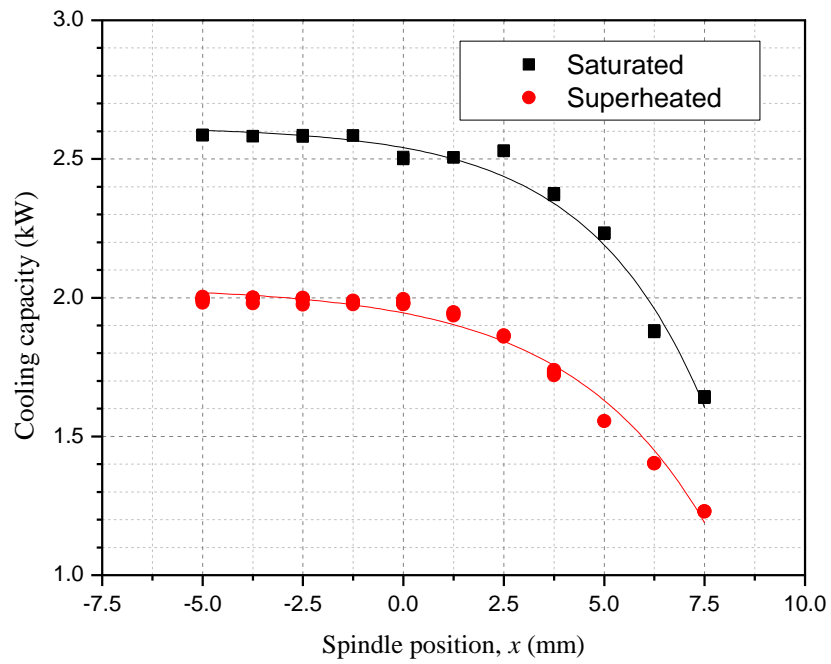
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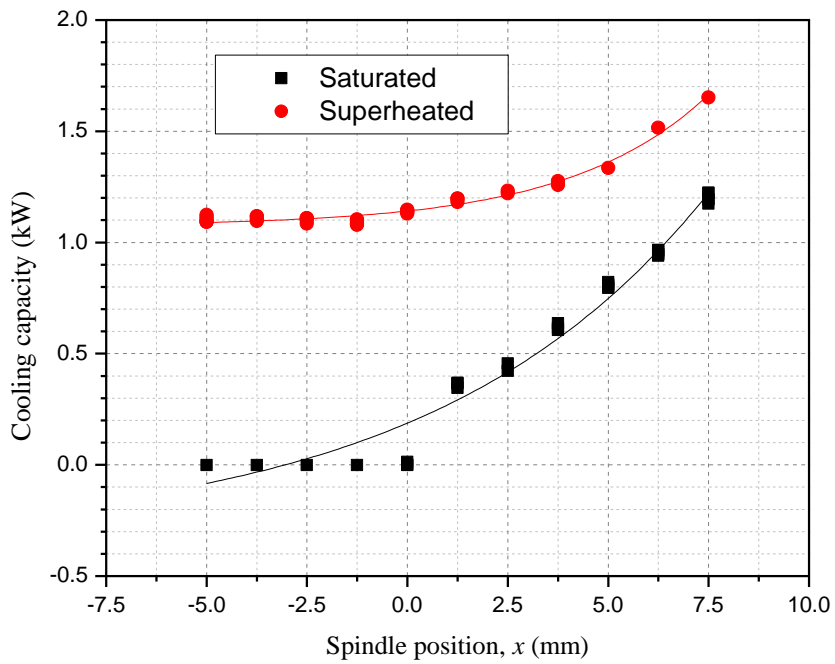
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(a)



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(b)

522 Fig. 7 Variations of cooling capacity with spindle position at saturated and superheated conditions

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(a) cooling capacity of Evaporator 1, (b) cooling capacity of Evaporator 2

524 **Tables**

525

526

527 Table 1 Test conditions to study effect of primary fluid state

528

	Label	Application	Pressure (kPa)	Temperature of ethylene glycol solution (°C)
High-temperature evaporator	Evaporator 1	Refrigeration	240	5~25
Low-temperature evaporator	Evaporator 2	Freezing	110	-5

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532 Table 2 Test conditions to study effect of ejector area ratio

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Test Parameter	Label	Value
Primary pressure, $p_P$ (kPa)	P5	240
Secondary pressure, $p_S$ (kPa)	P7	110
Temperature of Evaporator 1 at saturated condition (°C)	T9	2
Temperature of Evaporator 1 at superheated condition (°C)	T9	15
Spindle position, $d_{N1S1}$ (mm)		0 ~ 12.5
Area ratio		6.25~8.33

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