Experimental Determination of an Optimal Performance Map of a Steam Ejector Refrigeration System

Kittiwoot Sutthivirode 1,2 and Tongchana Thongtip 1,2, *  

1 Thermal and Fluid Laboratory (TFL), Department of Teacher Training in Mechanical Engineering, King Mongkut’s University of Technology North Bangkok, 1518 Phacharat 1 Rd., Bang Sue, Bangkok 10800, Thailand; kittiwoot.s@fte.kmutnb.ac.th  
2 Advanced Refrigeration and Air Conditioning Laboratory (ARAC), Department of Teacher Training in Mechanical Engineering, King Mongkut’s University of Technology North Bangkok, 1518 Phacharat 1 Rd., Bang Sue, Bangkok 10800, Thailand  
* Correspondence: tongchana.t@fte.kmutnb.ac.th; Tel.: +66-2-5552000

Abstract: An experimental determination of optimal performance of a steam ejector refrigerator was proposed which aims to indicate the optimal performance under various heat source temperatures. A small-scale steam ejector refrigerator test bench was constructed to carry out the experiment and to determine the optimal performance map. Three primary nozzles with throat diameters of 1.4, 1.6, and 1.8 mm, were tested with an ejector throat diameter of 14.5 mm, providing the ejector area ratios of 107, 82, and 65, respectively. For a particular working condition, the boiler temperature was varied to determine the maximum COP which is recognized as the optimal operation. It was found that the secondary fluid stream is first choked at the optimal boiler temperature. This optimal point varied significantly with the evaporator temperature, condenser pressure, and ejector area ratios. It was found that this steam ejector refrigerator could be operated under the optimal boiler temperature between 102.5 and 117.5 °C depending on the ejector area ratio, evaporator temperature, and condenser pressure. The optimal performance map is beneficial to further control the heat source temperature so that the maximum COP is achieved.

Keywords: steam ejector; ejector refrigeration system; performance map; heat driven refrigerator

1. Introduction

Using refrigeration machines to produce useful cooling production faces a problem of energy shortages in the future because most machines are electrically powered by vapor-compression systems. Therefore, many efforts to reduce the electricity consumption for the refrigeration process have been made by researchers [1–8]. The use of a heat-powered refrigeration machine is a promising solution to reduce the electricity consumption because it satisfies the cooling purposes, as well as providing a huge reduction in electrical energy consumption. Thus, the novel research based on the heat-powered refrigeration machine is a main focus in this research field. Thus, several researchers have proposed the novel heat-powered refrigerator as supported by Mahmoudian et al. [9], Eames et al. [10], Li et al. [11], Aidoun et al. [12], and Fang et al. [13].

A heat-powered ejector refrigeration system is a promising refrigeration machine due to the fact that it is quite simple to design, to construct, and to run as compared with other refrigeration machines. Additionally, the ejector refrigerator requires a quite low installation cost and maintenance cost under the same cooling capacity which is due to the fact that it has fewer moving parts which provides a longer lifetime. In addition, various synthetic refrigerants or even steam are able to be used as the working fluid to produce useful cooling production. Therefore, ejector refrigerators using various refrigerants such as HFC, HCFC, and HFO have been proposed by several researchers in an attempt to demonstrate the capability of the ejector refrigerator powered by low temperature,
heat sources. Ejector refrigerators working with different working fluids were implemented by Aphornratana et al. [14] (for R11), Ruangtrakoon et al. [15], Wang et al. [16] and Huang et al. [17] (for R141b), Yapici [18] and Ersoy et al. [19] (for R123). They found that it is possible to use the ejector refrigerator under a relatively low heat source temperature (below 100 °C). However, such synthetic refrigerants produce a relatively high global warming potential (GWP) and ozone depletion potential (ODP) which raises concerns about the environmental impact. This has encouraged a new generation of refrigerants, intended to mitigate environmental impact as well as produce reasonable performance, to be developed to replace the old refrigerants. Modern synthetic refrigerants such as HFO-245fa, HFO-1233zd (E), and HFO-365mfc have been proven to be promising refrigerants for working with the ejector refrigerator. This is due to their boiling point and thermodynamic properties as supported by Besagni et al. [20], Mahmoudian et al. [21], and Mazzelli et al. [22]. Even though the modern synthetic refrigerants have been proven to be suitable for ejector refrigerators in terms of performance and environmental impact, their cost and availability are major limitations for practical use in the ejector refrigerator. Because of this, the ejector refrigerator working with steam is still popular in this research field due to the fact that it is more practical and more economical. Another advantage of using steam as the working fluid over modern refrigerants is that the waste steam or steam tab from industrial boilers can be directly used to drive the ejector, while for the refrigerant, the heat exchanger is required to generate refrigerant vapor to operate the ejector. Thus, the practical use of the steam ejector refrigerator has been the main focus of several researchers [23–27]. The main aim is to improve the system COP or entrainment ratio. However, to improve the steam ejector refrigerator performance, better understanding of the flow condensation along the ejector and condensation shock wave are highly required. This has caused some researchers, Zhang et al. [3] and Zhang et al. [26], to investigate the flow condensation of the steam ejector based on the CFD technique and experiment for providing a clear explanation on the key to improving steam ejector performance. Hence, new knowledge on the steam ejector technology was proposed.

Ejector refrigerators have been proposed for real air conditioning applications as proposed by Thongtip et al. [23], Varga et al. [25], and Ruangtrakoon et al. [27]. The real air-conditioning test of the steam ejector refrigerator was proposed by Ruangtrakoon and Aphornratana [27] in a hot humid climate. Their steam ejector refrigerator was operated under a boiler temperature of 110–130 °C and a cooling capacity of 4.5 kW. However, the refrigeration machine can be used for air conditioning applications under a specific range of boiler temperatures. The existing research proposed by Ruangtrakoon and Thongtip [15] and Yapici [18] showed that variations in the boiler temperature significantly affected the system COP or mass entrainment ratio. Their findings were that, for specified evaporator temperature and condenser pressure, a particular ejector must be operated with an optimal boiler temperature. This is to achieve the maximum COP. Additionally, as the evaporator temperature and condenser pressure were varied, the optimal boiler temperature changed significantly. More interestingly, it was found that even if the ejector geometries were optimized carefully by CFD or experiment, the boiler temperature is still a key parameter of interest which needs to be studied carefully. These previous studies have indicated that the working temperature of the boiler plays a key role in the cooling performance even if the ejector geometries and primary nozzle are optimized precisely.

In practice, the ejector refrigerator must be run at the critical operating condition in which the maximum system COP is obtained, whilst the unwanted heat is rejected at the maximum condensation pressure. This critical operating condition is determined by the common performance curves in which the entrainment ratio (Rm) or system COP versus the condenser saturation pressure is illustrated. Therefore, an ejector performance assessment is extensively discussed using the common performance curve as proposed by [16,17,24]. Their aims were to indicate the performance improvement via geometrical optimization.

As the ejector refrigerator is operated under the real scenario, the condensation pressure is dominated by the heat sink (ambient condition). In this case, the working tem-
perature of the boiler and evaporator are recognized as the main factors to perform the critical point of operation. Hence, the working temperature of the boiler and evaporator has a strong influence on the critical operating condition. Even though the primary nozzle and ejector mixing chamber geometries are optimized carefully, optimal performance still changed with the evaporator and boiler temperature. Therefore, it is inconvenient to change the ejector and primary nozzle during the real operation. To this end, the determination of the proper value of the boiler and evaporator temperature is an alternative way to determine the critical operating condition which is easier.

At a specified working condition, if the boiler temperature is not high adequate to overcome the condensation pressure, the ejector refrigerator is unable to operate efficiently or may even malfunction. The reason is that the primary stream momentum is inadequate to produce the efficient entrainment process as explained by Zhu et al. [28], Ruangtrkoon et al. [29], and Ariafar et al. [30]. As a result, the secondary stream is not choked, producing a relatively low cooling capacity. However, using a too high boiler temperature also produces a lower system COP because a higher heat rate is needed for operation (a higher primary mass flow rate is allowed through the primary nozzle). Although the maximum cooling capacity is achieved because of the secondary fluid stream being choked, a higher amount of the primary fluid is the result of producing a lower system COP. This means that the excessive primary fluid flow rate does not give advantage to the ejector refrigerator operation. Therefore, the ejector refrigerator must be operated under the proper boiler temperature which is recognized as the optimal value. However, this optimal boiler temperature is not able to be determined directly during the real operation since the operating condition must be controlled precisely by the researchers. Hence, the experimental work must be conducted to determine the optimal point of operation. This is because all working conditions can be regulated precisely. Moreover, the experimental results provide a reference case to determine the optimal performance map of the ejector refrigerator. This is useful for further developing the ejector refrigerator driven by different heat source temperatures such as the hot water produced by an evacuated tube solar collector, steam tab from boiler, flue gas heat recovery, etc. However, from open literature, the optimal performance map based on the variations in the heat source temperature is still missing since the previous research focused on determining the critical operation with the common performance curve.

This present work proposes an experimental determination of an optimal performance of a steam ejector refrigerator which aims to demonstrate the optimal working condition under various boiler temperatures (various heat source temperatures). A steam ejector refrigerator test bench was built for experiments. Three primary nozzles with different throat diameters of 1.4, 1.6, and 1.8 mm (but the same nozzle area ratio), were equipped with an ejector throat diameter of 14.5 mm associated with ejector area ratios of 107, 82, and 65, respectively. At a specified working condition, the boiler temperature was varied which aims to determine the maximum COP. Hence, it is recognized as the optimal value of the boiler temperature. It was found that the secondary fluid stream is first choked at the optimal boiler temperature. The performance is also determined under various evaporator temperatures, condensation pressures, and primary nozzles to produce the optimal performance map. The optimal performance of this steam ejector refrigerator showed that it can be operated under a boiler temperature between 102.5 and 117.5 °C (under the maximum COP of 0.56 and 0.36) depending on the ejector area ratio, evaporator temperature, and condensation pressure. The optimal performance map determined in this present work is beneficial for further controlling the heat source temperature so that the maximum COP is achieved.

2. Experimental Apparatus

2.1. A Steam Ejector Refrigerator Test Bench

A steam ejector refrigerator test bench was designed and constructed for experiments. It was designed in an attempt to investigate the parameter of interest accurately. Hence,
the cooling performance of the steam ejector refrigerator with varying working conditions and ejector geometries will be demonstrated and its optimal performance will later be determined. The picture and schematic view of the steam ejector refrigerator test bench are shown in Figure 1. The major components include a boiler (1), an evaporator (2), an ejector (3), a condenser (4), a receiver tank (5), a feed pump (6), and measuring devices.

**Figure 1.** The picture and schematic view of a steam ejector refrigerator test bench.

A boiler’s vessel was made of 5-inch stainless steel pipe (SUS-304, schedule 40 s) with a length of 1.5 m. The vessel was sealed by Teflon joint sealant. A heater with a rated power up to 8.0 kW was installed to produce steam under the desired boiler temperature. Along the top part of the boiler, the baffle plates are employed for separating the liquid droplets being carried over with steam. The water level within the boiler was monitored by the sight glass. Therefore, the decreasing rate of water over a certain time interval was measured directly. Hence, the mass flow rate of the primary stream was determined. The boiler saturation temperature value was measured by a thermocouple (k-type) while the desired temperature was obtained precisely by a temperature controller. The working pressure of the boiler vessel was indicated by a pressure gauge and pressure transducer. For safety, a pressure relief valve was employed to limit the working pressure of the boiler.

An evaporator’s vessel was modified from 3-inch stainless steel pipe (SUS-304, schedule 10 s) with a length of 80 cm. A heater with a rated power up to 2 kW was installed to produce the cooling load. The spray falling film technique was applied to enhance the evaporation process of the water within the evaporator. The saturation temperature was measured by thermocouple (k-type) and the desired temperature was achieved precisely
via a thermostat. The neoprene foam insulator was applied to prevent unwanted heat gain to the evaporator. The water level within the evaporator’s vessel was observed by a sight glass which allowed the dropping rate of liquid water over a certain time interval to be recorded.

The condenser was a water-cooled type. The cooling water temperature was controlled precisely by a water chiller (vapor compression chiller with cooling capacity of 7.0 kW). During the experiment, the desired condensation pressure for a certain working condition was obtained by regulating the cooling water temperature and its volume flow rate. The condensate from the condenser was stored within the receiver tank. It was fed to the boiler and evaporator by a magnetic-coupling gear pump. The condenser saturation pressure was indicated by an absolute pressure transducer. Four ball–valves and two manifolds were installed to control the liquid level within the boiler and evaporator.

2.2. The Ejector Geometries

In this present work, one particular ejector was studied with three primary nozzles. Their primary nozzle throat and exit diameter were varied aiming to provide the same area ratios (same nozzle exit Mach number). The ejector component was designed associated with the improved 1-D theory proposed by Kitrattana et al. [31] which aims to produce a nominal cooling capacity of 1.0 kW. The ejector geometries are presented in Figure 2 and Table 1.

![Figure 2](image_url)

Figure 2. The dimension of the ejector and primary nozzle.

<table>
<thead>
<tr>
<th>Table 1. The primary nozzle dimension.</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Nozzles</strong></td>
</tr>
<tr>
<td>------------</td>
</tr>
<tr>
<td>D1.4</td>
</tr>
<tr>
<td>D1.6</td>
</tr>
<tr>
<td>D1.8</td>
</tr>
</tbody>
</table>

The material used to construct the primary nozzle, mixing chamber, throat, and diffuser was brass. It was manufactured via an electrical discharging machine (EDM). The primary nozzle was the converging–diverging type for achieving the supersonic speed at the nozzle outlet. The conical converging section was based on a converging radius of 7 mm which aims to minimize the friction loss. The diverging section was constructed with a half angle of 4.5° to avoid the separation and recirculation flow. Three primary nozzles were manufactured with the uncertainty of the primary nozzle throat and exit diameter of...
±0.05 mm. The primary nozzle exit position (NXP) was placed inside the mixing chamber at NXP = +20 mm.

2.3. Data Reduction and Instrumentation

During the experiments, the cooling performance of the steam ejector refrigerator performance will be represented by the system COP. The system COP is determined when the relevant parameters including temperature, pressure, mass flow rate, and electrical energy are obtained. The system COP can be determined by Equation (1),

\[
COP = \frac{\dot{Q}_{\text{evap}}}{\dot{Q}_{\text{boiler}} + W_{\text{pump}}}
\]

where \(\dot{Q}_{\text{evap}}\) is the cooling load produced at the evaporator (W), \(\dot{Q}_{\text{boiler}}\) is the heat required for boiler operation (W), \(W_{\text{pump}}\) is the work required for the liquid pump (W).

The cooling load at the evaporator can be determined by Equation (2),

\[
\dot{Q}_{\text{evap}} = \dot{m}_s (h_{g@T_{\text{evap}}} - h_{f@T_{\text{cond}}})
\]

where \(\dot{m}_s\) is the secondary mass flow rate (kg/s), \(h_{g@T_{\text{evap}}}\) is the saturated vapor specific enthalpy at the evaporator outlet, \(h_{f@T_{\text{cond}}}\) is the saturated liquid specific enthalpy at the condenser outlet.

The heat rate required for operating the boiler is calculated by Equation (3),

\[
\dot{Q}_{\text{boiler}} = \dot{m}_p (h_{g@T_{\text{boiler}}} - h_{f@T_{\text{cond}}})
\]

where \(\dot{m}_p\) is the primary mass flow rate (kg/s), \(h_{g@T_{\text{boiler}}}\) is the saturated vapor specific enthalpy at the boiler outlet, \(h_{f@T_{\text{cond}}}\) is the saturated liquid specific enthalpy at the condenser outlet.

The electrical energy required for all liquid pumps can be calculated by Equation (4),

\[
W_{\text{pump}} = VI \cos \phi
\]

where \(V\) is the voltage of the Alternating Current (Volt), \(I\) is the current (A), \(\cos \phi\) is the power factor of the Alternating Current.

For a certain working condition, the steam ejector refrigerator performance is represented by COP against the boiler temperature’s variation. Thus, the temperature, pressure, flow rate, and electrical energy must be measured accurately. In this case, the measuring devices and experimental technique must be reliable. Type-k thermocouple probes were calibrated carefully before implementing the experiment. From calibration, the uncertainty of the thermocouple probe is ±0.2 °C. A digital temperature controller operated with solid state relay was used to obtain the desired temperature. The pressure transducers were calibrated carefully. From calibration, the uncertainty of the pressure transducers was about ±1.0% of full scale.

The primary mass flow rate (\(\dot{m}_p\)) and secondary mass flow rate (\(\dot{m}_s\)) were determined by measuring the dropping rate of the liquid water level within the boiler and evaporator, respectively. Herein, the water level disappearing from the boiler and evaporator under a certain time interval was monitored. The uncertainty of the stopwatch used was ±0.01 s while that of measuring the liquid level was ±0.1 mm. From uncertainty analysis, the uncertainty of the primary and secondary mass flow rate was ±2.25% of read values. Thus, the uncertainty of the system COP was ±2.85% of read values. During the operation, the steady state operation of the system was indicated by data acquisition (Yokokawa, model GP10-1-E-F/UC20), making the operation of the steam ejector refrigerator and the recording of results reliable.
3. Results and Discussion

This present work aims to experimentally demonstrate the performance of the steam ejector refrigeration system when working under various heat source temperatures. The reference case for further developing the steam ejector refrigerator driven by available heat sources such as solar-water heater-based evacuated tube collector, exhaust gas from combustion, steam tab from industrial boiler, etc. Therefore, this present work will concentrate on how the boiler temperatures variation affects the cooling performance. As mentioned earlier, a certain value of the boiler temperature at which the ejector can perform the maximum COP will be determined experimentally. The reason is that the momentum transfer process is created by the boiler temperature. Therefore, the optimal boiler temperature must be determined under the actual operation. In real operation of the ejector refrigerator, the condensation temperature depends on the ambient condition. Hence, the condenser saturation temperature is held constant for a particular working condition. In addition, the evaporator temperature is also kept constant. This will indicate the variations in the cooling load applied to the evaporator for achieving the constant temperature.

During the experiments, the boiler temperature will be varied from 98 to 130 °C which is associated with the available heat source. Additionally, the evaporating temperature will range from 4 to 12 °C which is consistent with an air-conditioning application. The condensation temperature will range from 29–32 °C in which the cooling water can be produced by the common cooling tower in Thailand’s climate. The impact of the aforementioned working condition on the overall performance of the steam ejector refrigerator will be discussed and the optimal performance map will be determined for an actual reference case.

3.1. Impact of the Boiler Temperature on the System Performance

To investigate this impact, the evaporator temperature was kept constant at 8 °C. The condensation pressure was also held constant at 4200 Pa corresponding to the condensation temperature of 29.8 °C. A primary nozzle, D1.6, was used for investigation. The boiler temperature was increased from 98 to 130 °C. The primary fluid and secondary fluid flow rate were measured and the cooling load, heat rate required, system COP were later determined for discussion. The experimental results are shown in Figures 3 and 4.

It is seen from Figure 3a that increasing the boiler temperature causes the cooling load to vary significantly. Initially, the cooling load increases with increasing the boiler temperature until its maximum value is reached. Later, the cooling load slightly decreases with increasing the boiler temperature. However, the heat rate for operating the boiler is increased as the boiler temperature increases, as shown in Figure 3a, due to producing a higher primary mass flow rate. As a result, the system COP initially increases with increasing the boiler temperature until it reaches its maximum value. Later, it decreases with increasing the boiler temperature as seen in Figure 3b. Hence, the boiler temperature value at which the system COP reaches its maximum value can be considered as the optimal value for this specified working condition. It is obvious that this tested method can indicate the optimal heat source temperature for the steam ejector refrigerator. For this particular test case, the optimal boiler temperature is 112.5 °C which produces a maximum COP of 0.42.

The reason why the system COP is quite low when operating the boiler temperature below the optimal value is that the secondary mass flow rate produced by the evaporator is relatively low. In this case, it is thought that the secondary fluid stream does not reach sonic speed; thus, the choked flow of the secondary fluid stream is not possible. In this case, the primary stream momentum is inadequate to perform the choked flow of the secondary fluid stream. This implies that when the boiler temperature reaches its optimal value, the momentum of the primary fluid stream is high enough for efficiently operating the steam ejector.
Figure 3. Parameters of interest influenced by the boiler temperature. (a) The rated power, thermal and electrical energy, with variations in the boiler temperature; (b) Variations in the COP with the boiler temperature; and (c) Variations in the entrainment ratio with the boiler temperature.
**The optimal boiler temperature (optimal operation), * the critical condensation pressure (critical operation).

while the boiler temperature is kept constant at 112.5 °C with the critical operation.

particular ejector is 4200 Pa. It is obviously seen that the critical working condition (point a)

pressure is equal to or lower than 4200 Pa. Thus, the critical condenser pressure of this

Comparison of the performance of the steam ejector refrigerator at the optimal operation

Table 2. Comparison of the performance of the steam ejector refrigerator at the optimal operation

<table>
<thead>
<tr>
<th>Point</th>
<th>T_{boiler} (°C)</th>
<th>T_{evap} (°C)</th>
<th>P_{cond} (Pa)</th>
<th>( \dot{m}_s )</th>
<th>( \dot{m}_p )</th>
<th>COP</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>112.5 **</td>
<td>8</td>
<td>4200</td>
<td>0.00037</td>
<td>0.00089</td>
<td>0.42</td>
</tr>
<tr>
<td>a</td>
<td>112.5</td>
<td>8</td>
<td>4250 *</td>
<td>0.00038</td>
<td>0.00087</td>
<td>0.42</td>
</tr>
</tbody>
</table>

** The optimal boiler temperature (optimal operation), * the critical condensation pressure (critical operation).

From the above comparison, it can be ensured that the first choked flow of the secondary fluid stream is found when the ejector refrigerator is run at the optimal boiler temperature. Comparing the results obtained from Figure 4 with those obtained from

Figure 4. The conventional performance curve.

When operating the boiler temperature above the optimal value, a slight decrease in

the cooling load is found while the primary fluid mass flow rate is increased significantly.

In this case, the secondary fluid stream is still choked but the secondary mass flow rate

decreases because the flow area for the entrained secondary fluid is reduced. In such a case,

a higher primary mass flow rate is produced with increasing the boiler temperature, and,

thus, it must occupy a larger flow area within the ejector mixing chamber. Thus, the flow

area for the entrained secondary fluid is reduced which results in a lower secondary mass

flow rate.

It is obviously seen that when the particular ejector is operated at the optimal boiler
temperature, the secondary fluid stream is first choked. However, this must be proved to

ensure that the secondary fluid is really choked at the optimal boiler temperature. To prove

this, the conventional performance curve when the boiler temperature is kept constant at

the optimal value (determined from Figure 3) is depicted as shown in Figure 4.

It is seen from Figure 4 that the evaporator temperature is held constant at 8 °C

while the boiler temperature is kept constant at 112.5 °C (the optimal boiler temperature
determined from Figure 3). The condenser pressure is increased until the system COP

drops to zero. Figure 4 shows that the system COP is approximately maintained at the

maximum value of 0.42 (due to working on the choked flow region) when the condenser

pressure is equal to or lower than 4200 Pa. Thus, the critical condenser pressure of this

particular ejector is 4200 Pa. It is obviously seen that the critical working condition (point a

in Figure 4) is identical to the optimal working condition (point A in Figure 3). The results

of two working conditions are summarized in Table 2.
Figure 3, the choked flow of the secondary stream begins when operating the ejector at the critical condenser pressure of 4200 Pa which is identical to the proposed performance curve as depicted in Figure 4. This means that if the boiler temperature is lower than the optimal value, the primary fluid momentum is inadequate to perform the choked flow of the secondary fluid stream. In such a case, the ejector refrigerator will be operated on the unchoked flow region as depicted in Figure 4 (indicated by the green dashed line). Herein, the momentum of the primary fluid stream is inadequate to overcome the condenser pressure and, thus, the secondary fluid stream cannot reach the sonic speed.

It is also evident from Figure 3 that, throughout the unchoked flow region, an increase in the boiler temperature causes production of a higher secondary fluid mass flow rate. Thus, a higher COP or entrainment ratio increases with increasing the boiler temperature when the boiler temperature is lower than its optimal value. At the boiler temperature above the optimal point, the COP or entrainment ratio decreases with increasing the boiler. It is noticeable that even if the secondary flow is choked (as indicated by the red dashed line in Figure 4), a lower secondary mass flow rate is obtained. Therefore, it is ensured that when the boiler temperature is higher than the optimal value, the ejector refrigerator will always operate on the choked flow region.

As discussed above, under a fixed evaporator temperature and condenser saturation pressure, the steam ejector refrigerator should be operated with the optimal boiler temperature. This is so that its best system performance can be achieved. The results show that using a too high boiler temperature is useless for operating the steam ejector refrigerator. However, the optimal boiler temperature is believed to be changed significantly with variation in the condenser pressure and evaporator temperature. Their effects will be investigated in the next section.

3.2. Impact of the Evaporator Temperature on the Optimal Performance

To investigate this, the condenser was held constant at 4200 Pa (condensation temperature of 29.8 °C) and the evaporator temperature was ranged from 4 to 12 °C. For a fixed evaporator temperature, the boiler temperature was increased from 100 to 130 °C for determining the optimal boiler temperature. The primary nozzle, D1.6, was studied. The results are shown in Figures 5 and 6.

Figure 5. The system COP influenced by the evaporator.
It is found from Figure 5 that as the evaporator temperature is increased from 4 to 12 °C, the steam ejector refrigerator is able to operate with a lower optimal boiler temperature. In addition, the system COP at the optimal point increases with increases in the evaporator temperature. The optimal boiler temperatures for evaporator temperatures of 4, 8, and 12 °C are 117.5, 112.5, and 107.5 °C, respectively. Meanwhile, the maximum COP at the evaporator temperatures of 4, 8, and 12 are 0.36, 0.42, and 0.56, respectively. From the results, the optimal boiler temperature under various evaporator temperatures must be determined for being a reference case to reflect the real operation of the steam ejector. This is useful in the case of using the steam ejector refrigerator to produce different cooling temperatures. The results also indicate that the boiler temperature must vary significantly with the cooling temperature (evaporator temperature) to obtain the maximum COP. These results are useful for controlling the heat source temperature and heat rate supplied to the boiler for developing the steam ejector refrigeration system driven by real heat sources.

The reason why the steam ejector refrigerator can be operated with a lower optimal boiler temperature when raising the evaporator temperature is that the saturation evaporator pressure is higher when raising evaporator temperature. This causes the mixing chamber pressure (associated with the evaporator pressure) to be higher. Hence, for a certain condenser pressure, the secondary fluid stream begins to be choked at a lower boiler temperature. Therefore, the optimal performance of the steam ejector refrigerator can be achieved at a lower boiler temperature. In addition, at the higher upstream pressure, a higher secondary fluid flow rate is produced within the evaporator and is later drawn into the mixing chamber. As a result, a higher cooling load and system COP are achieved.

To further prove the first choke flow of the secondary stream working with the optimal boiler temperature under different evaporator temperatures, the common performance curve is also determined under different evaporator temperatures while fixing the boiler temperature at the optimal value (determined from Figure 5). Therefore, the boiler temperature is kept constant at the optimal values for determining the conventional performance curve which are as follows: fixing $T_{\text{boiler}} = 117.5 \, ^\circ\text{C}$ at $T_{\text{evap}} = 4 \, ^\circ\text{C}$; fixing $T_{\text{boiler}} = 112.5 \, ^\circ\text{C}$ at $T_{\text{evap}} = 8 \, ^\circ\text{C}$; and fixing $T_{\text{boiler}} = 107.5 \, ^\circ\text{C}$ at $T_{\text{evap}} = 12 \, ^\circ\text{C}$. The results based on the common performance curve are illustrated in Figure 6.

From Figure 6, when fixing the boiler temperature at the optimal value, the critical condenser pressure produced at the evaporator temperatures of 4, 8, and 12 °C (points A, B, and C) are similar which is 4200 Pa (condensation temperature of 29.7 °C). The maximum COPs obtained at evaporator temperatures of 4, 8, and 12 °C are 0.36, 0.42, and 0.56. These
results are identical to those shown in Figure 5 (points a, b, and c) when working with the optimal boiler temperature. The comparison of the results based on the critical operation (Figure 6) and those based on the optimal operation (Figure 5) are tabulated in Table 3.

Table 3. The optimal and critical performance of the steam ejector refrigerator.

<table>
<thead>
<tr>
<th>Point</th>
<th>$T_{\text{boiler}}$ (°C)</th>
<th>$T_{\text{evap}}$ (°C)</th>
<th>$P_{\text{cond}}$ (Pa)</th>
<th>%P$_{\text{cri-diff}}$</th>
<th>COP</th>
<th>%COP$_{\text{cri-diff}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>107.5 **</td>
<td>12</td>
<td>4200</td>
<td>-</td>
<td>0.560</td>
<td>0.88%</td>
</tr>
<tr>
<td>a</td>
<td>107.5</td>
<td>12</td>
<td>4200 *</td>
<td>-</td>
<td>0.565</td>
<td>0.88%</td>
</tr>
<tr>
<td>B</td>
<td>112.5 **</td>
<td>8</td>
<td>4215</td>
<td>0.35%</td>
<td>0.420</td>
<td>0.47%</td>
</tr>
<tr>
<td>b</td>
<td>112.5</td>
<td>8</td>
<td>4200 *</td>
<td>0.35%</td>
<td>0.422</td>
<td>0.47%</td>
</tr>
<tr>
<td>C</td>
<td>117.5 **</td>
<td>4</td>
<td>4200</td>
<td>0.47%</td>
<td>0.361</td>
<td>0.83%</td>
</tr>
<tr>
<td>c</td>
<td>117.5</td>
<td>4</td>
<td>4220 *</td>
<td>0.47%</td>
<td>0.358</td>
<td>0.83%</td>
</tr>
</tbody>
</table>

** The optimal boiler temperature; * the critical condenser pressure from the conventional performance curve; %P$_{\text{cri-diff}}$ is the percent difference between the critical condenser pressure and the condenser pressure at the optimal operation. %COP$_{\text{cri-diff}}$ is the percent difference between COP achieved at the critical point and that achieved from the optimal operation.

As discussed above, it was proven that the critical working condition based on the common performance curve under various evaporator temperatures is similar to that based on the optimal working condition. As previously discussed in Section 3.1, it showed that the secondary stream begins to be choked when the refrigerator is operated with the optimal boiler temperature. Therefore, in this section, the first choked flow of the secondary stream under various evaporator temperatures was also proven. This ensures that the secondary stream begins to be choked when working with the optimal boiler temperature.

As discussed in this section, the boiler temperature must essentially be varied with the cooling temperature (or evaporator temperature) for achieving the maximum COP. The results proposed in this section provide the guideline for tuning the boiler temperature under various evaporator temperatures. This is valuable because the results are determined based on the experiments which represents the practical performance of the steam ejector refrigerator. However, for providing more comprehensive optimal performance, the variations in the condenser pressure will be studied in the next section.

3.3. Impact of the Condenser Pressure’s Variation on the Optimal Performance

To investigate this impact, the primary nozzle, D1.6, was used for the experiment. The evaporator temperature was fixed at 8 °C. Meanwhile, the condenser pressure was fixed at different values ranging from 4200 to 5000 Pa ($T_{\text{cond}}$ between 29.8 and 34.2 °C). The boiler temperature was increased to determine the optimal operation. The system COPs were recorded for discussion.

From Figure 7, when raising the condenser pressure, the steam ejector refrigerator must be operated with a higher optimal boiler temperature for achieving the maximum COP. This is due to the fact that the compression shock wave takes place closer to the upstream of the mixing chamber due to being operated with a higher condensation pressure (higher ejector back pressure). This phenomenon was well documented by Ruangtrakoon et al. [27] and Zhu et al. [28]. Therefore, the mixing process is disturbed due to the impact of the shock wave, and, therefore, the secondary stream is unable to perform the choked flow at quite low boiler temperature. In this case, a higher boiler temperature is required to perform the first choked flow of the secondary stream.

Commonly, increasing the boiler temperature always yields a lower system COP because of producing a higher primary mass flow rate (using a higher heat rate for operation). However, the system COP produced at the optimal operation is the highest value. This can ensure that the steam ejector refrigerator always produces its best performance for a particular condensation pressure.
3.4. Impact of Primary Nozzle Throat (or Ejector Area Ratio) on the Optimal Performance

In these experiments, three primary nozzles, D1.4, D1.6, and D1.8, were studied. They were tested with one fixed ejector geometry with throat diameters of 14.5 mm; hence, the ejector area ratios (the ratio of the cross-sectional area of the ejector throat to the primary nozzle throat) were 107, 82, and 65, respectively. The evaporator temperature was fixed at 8 °C while the condensation pressure was kept constant at 4200 kPa (condensation temperature of 29.8 °C). The boiler temperature was raised continuously to determine its optimal value.

When the ejector is equipped with different primary nozzle throat sizes, the ejector area ratio (AR$_{ej}$) is changed. This ejector area ratio is a significant dimensionless parameter which dominates the ejector performance. It has a great impact on ability to produce the critical entrainment performance. From the published works by Pereira et al. [32], Wan et al. [33], under the common performance curve (COP versus P$_{cond}$), an ejector with a higher AR$_{ej}$ provided a higher Rm and COP. However, it produced a lower critical condenser pressure. It was evident that if the ejector geometries are enlarged proportionally to provide the same AR$_{ej}$, the ejector always produced the same Rm and critical condensation pressure when operating with the same working temperature of the boiler and evaporator. However, there is one difference which is that a bigger ejector throat produced a higher cooling capacity. Herein, the ejector performance when working with different primary nozzle throat sizes is assessed associated with the ejector area ratio. This will provide a baseline for designing the ejector to be consistent with the desired cooling load (when scaling-up or scaling-down the steam ejector refrigerator).

Figure 7 depicts the influence of the ejector area ratio on the optimal performance. It is found that when using a larger AR$_{ej}$ (smaller nozzle throat), the steam ejector refrigerator must require a higher optimal boiler temperature for achieving the maximum COP. Even if the optimal boiler temperature is increased when using a larger AR$_{ej}$, it produces a higher system COP. This implies that a higher boiler temperature is required for performing the choking of the secondary fluid stream. This is because using a smaller primary nozzle throat (larger AR$_{ej}$), a lower primary mass flow rate is produced while the condenser pressure is fixed. Herein, a higher primary stream momentum is needed to overcome the condenser saturation pressure. This is so that the secondary fluid stream begins to be choked. Therefore, the ejector is essentially operated with a higher boiler temperature for achieving the optimal performance.
As proposed above, under the ejector area ratios of between 65 and 107, the optimal performance is produced under the boiler temperature between 105 and 118 °C, respectively. Such boiler temperature values are relatively low which is consistent with the available heat sources such as steam tab from the industrial boiler, hot water produced by evacuated tube solar collectors or parabolic trough solar collectors. The proposed results indicate the high potential of the steam ejector refrigerator for practical use driven by real heat sources. In addition, the condensation temperature demonstrated in this section is quite high (around 29.7 °C, consistent with the condenser pressure of 4200 Pa). This indicates the possibility of using the particular ejector area ratios under a quite high ambient temperature while producing a reasonable system COP. This finding may be used for further development of steam ejector refrigerators for practical use.

3.5. Optimal Performance Map of the Steam Ejector Refrigerator

As discussed in Sections 3.1–3.4, for particular ejector geometries, the steam ejector refrigerator requires an optimal boiler temperature to achieve a maximum system COP. In addition, the optimal boiler temperature varies significantly with the evaporator temperature and condensation pressure. This indicates that the steam ejector refrigerator must always be operated with the optimal boiler temperature for efficiently producing the cooling performance. Thus, the optimal performance map, which represents the overall working condition, is required for convenient development of the real ejector refrigeration plant. Moreover, in the case of using the solar water heater or steam tab from an industrial boiler to drive the ejector refrigerator, there is frequent fluctuation in the heat source temperature. Herein, the optimal performance can be determined by means of precisely controlling the boiler temperature. This is made possible when controlling the heat rate supplied to the boiler. However, the performance map of the steam ejector refrigerator is essentially established by experiments. This is due to the fact that the operating conditions can be regulated precisely to determine the optimal performance. In this section, the optimal performance map of the steam ejector refrigerator is proposed as a reference case. The optimal boiler temperature for a specified evaporator temperature and condenser pressure is determined based on the method previously provided in Sections 3.1–3.4. The optimal performance map is shown in Figure 9. This optimal performance map (based on the heat source temperature) is first proposed in this present work.
Figure 9. The optimal performance map under various working conditions and the ejector area ratios.

Figure 9 depicts the optimal working operation of the steam ejector refrigerator. The optimal boiler temperature under various evaporator temperatures, condenser pressures, and ejector area ratios is demonstrated simultaneously. Hence, this map can directly be
used to estimate the optimal working condition efficiently because it is determined based on the experiments as proposed in Sections 3.1–3.4. Therefore, the optimal boiler temperature for the steam ejector refrigerator operation can be determined easily and correctly. This makes it easier to further develop the steam ejector refrigerator driven by the heat source with frequent temperature fluctuations.

From Figure 9, with different evaporator temperatures and the condenser pressures, there are three working areas (gray, red, and blue) depending on the ejector area ratios. Each working area indicates the optimal performance map for certain ejector area ratios: $AR_{ej} = 65$ (Figure 9a); $AR_{ej} = 82$ (Figure 9b); and $AR_{ej} = 107$ (Figure 9c). Additionally, each area has different working condenser pressures and evaporator temperatures. It is seen that the steam ejector refrigerator can be operated under the optimal boiler temperature as low as 102 °C at the condensation temperature of 28.9 °C and ejector area ratio of 65. This is consistent with the steam ejector refrigerator powered by a solar evacuated tube collector in a warm climate (such as Thailand). In such a case, the cooling water of around 27–34 °C is produced for operating the refrigerator as supported by Ruangtrakoon and Thongtip [23] and Thongtip and Aphornratana [34].

The optimal performance map also indicates that the highest optimal boiler temperature determined in this present work is 130 °C which is consistent with the heat source available from steam tab from an industrial boiler, steam tab from a steam turbine power plant, parabolic trough solar collector, etc. Such heat sources are commonly available and can be directly used to drive the steam ejector refrigerator. Therefore, the optimal performance map determined in this present work is beneficial for efficiently selecting the optimal boiler temperature for the particular ejector area ratio, cooling temperature, and condensation temperature.

4. Conclusions

The optimal performance of the steam ejector refrigerator under various heat source temperatures was investigated experimentally. The optimal boiler temperature (optimal heat source) for the particular ejector geometries was determined under various evaporator temperatures, condensation pressures, and ejector area ratios. It was evident that the optimal performance varied significantly with the change in working conditions. Hence, the optimal performance map, which represented the comprehensive performance of the steam ejector refrigerator, was proposed as a baseline to conveniently develop the ejector refrigerator associated with the desired cooling capacity and real heat source. The major findings of this present work could also be summarized as follows:

- At fixed working condition and $AR_{ej}$, there is a certain boiler temperature value which produced a maximum system COP. It is recognized as the optimal boiler temperature.
- The secondary fluid stream begins to be choked when the steam ejector is being operated at the optimal boiler temperature. It was also shown that the optimal working condition was identical to the critical working condition based on the common performance curve.
- For a certain condenser pressure and ejector area ratio, the optimal boiler temperature decreased when the evaporator temperature increased. The optimal system COP was increased when the evaporator temperature increased.
- For a certain evaporator and ejector area ratio, when the condenser pressure increased, the refrigerator must be run with a higher optimal boiler temperature. The optimal system COP was decreased when the condenser pressure increased.
- For a fixed evaporator and condenser pressure, the ejector which has a larger $AR_{ej}$ required a higher optimal boiler temperature to produce a maximum system COP. However, the optimal system COP increased when using a larger ejector area ratio.

From all experiments, the optimal performance map was determined for efficiently being the reference case. The map demonstrates the optimal performance under the specified ejector area ratio, evaporator temperature and condenser pressure. This map can help the researchers who need to develop the steam ejector refrigerator to be consistent
with the cooling capacity which aimed to produce the maximum COP. Hopefully, the experimental evidence proposed in this present work is useful for researchers in this research area.

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**Nomenclature**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
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<tbody>
<tr>
<td>AR</td>
<td>Area ratio (m²)</td>
</tr>
<tr>
<td>CFD</td>
<td>Computational fluid dynamics</td>
</tr>
<tr>
<td>COP</td>
<td>Coefficient of Performance</td>
</tr>
<tr>
<td>D</td>
<td>Diameters (mm)</td>
</tr>
<tr>
<td>EDM</td>
<td>Electrical discharging machine</td>
</tr>
<tr>
<td>GWP</td>
<td>Global warming potential</td>
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<tr>
<td>HCFC</td>
<td>Hydrochlorofluorocarbons</td>
</tr>
<tr>
<td>h</td>
<td>Specific enthalpy (kJ kg⁻¹)</td>
</tr>
<tr>
<td>I</td>
<td>Current (Ampere)</td>
</tr>
<tr>
<td>NXP</td>
<td>Nozzle Exit Position (mm)</td>
</tr>
<tr>
<td>ODP</td>
<td>Ozone depletion potential</td>
</tr>
<tr>
<td>P</td>
<td>Pressure (Pa)</td>
</tr>
<tr>
<td>PID</td>
<td>Proportional integral derivative control</td>
</tr>
<tr>
<td>R</td>
<td>Refrigerant</td>
</tr>
<tr>
<td>SUS</td>
<td>Stainless steel</td>
</tr>
<tr>
<td>T</td>
<td>Temperature (°C)</td>
</tr>
<tr>
<td>V</td>
<td>Voltage of the Alternating Current (Voltage)</td>
</tr>
<tr>
<td>m</td>
<td>Mass flow rate (kg s⁻¹)</td>
</tr>
<tr>
<td>Q</td>
<td>Thermal energy (W, kW)</td>
</tr>
<tr>
<td>W</td>
<td>Required work (W, kW)</td>
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</tbody>
</table>

**Subscripts**

- boiler: condition for the boiler
- ej: ejector
- evap: condition for the evaporator
- f: liquid
- g: vapor
- pump: condition for the pump
- p: the primary fluid condition
- s: the secondary fluid condition
- boiler: temperature at the boiler
- con: temperature at the condenser
- evap: temperature at the evaporator
- sat: saturated state
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