A thermodynamic investigation and optimization of an ejector refrigeration system using R1233zd(E) as a working fluid

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Abstract. The increasing contribution of heating, ventilation, air conditioning and refrigeration systems to energy consumption and global warming is driving the search for technologies that are energy efficient, clean and renewable. The ejector refrigeration system is one such system with the potential to reduce energy consumption and CO₂ emissions. It is simple, low cost, has no moving parts, and can use low grade heat sources such as solar or waste heat. However, the coefficients of performance (COP) of these systems are still low. Moreover, most studies on ejector refrigeration systems have used refrigerants that are not environmentally benign. In this paper, the performance of an ejector refrigeration system working with R1233zd(E) is numerically investigated. R1233zd(E) is a newly introduced Hydrofluoroolefin refrigerant with no ozone depletion potential and very low global warming potential. No studies on the performance of this refrigerant in ejector systems have been conducted. A novel mathematical model that uses ejector coefficients which are dependent on the evaporator and generator pressures to accurately determine performance was used in the present study. In the analysis, area ratios between 6.44 and 10.94, evaporator temperatures between 0 and 16°C, and generator temperatures between 70 and 110°C were considered. Results show that system performance in the critical mode of operation increases as the generator temperature reduces and as the evaporator temperature increases. Furthermore, there is an optimal generator temperature at each condensing and evaporator temperature with the highest COP. Correlations for the optimal generator temperature and optimal COP have been derived and presented.

1. Introduction

The increasing concerns of climate change due to global warming have led to a rise in the development and adoption of clean and renewable energy technologies. Buildings are among the largest energy users accounting for nearly 40% of the total energy demand in developed countries [1]. Space heating and cooling is one of the largest contributors to the energy used in buildings. Since most of the energy used today comes from fossil fuels, this leads to increased emission of harmful gases to the environment and subsequently leads to global warming and climate change. In addition to reducing energy use in conventional space heating and cooling technologies, there are several efforts to replace fossil fuel-based technologies with ones using clean and renewable energy systems.

Among technologies with the potential to reduce emissions from heating, ventilation and air conditioning (HVAC) applications is the ejector refrigeration system. The ejector system works with minimum energy consumption since the compressor in conventional vapor compression systems is replaced by a pump. Moreover, since there are no moving parts in the ejector refrigeration system, it is simple, low cost and requires less maintenance compared to other systems [2]. Moreover, ejector refrigeration systems are thermally activated, presenting an opportunity to use low grade energy such as solar or waste heat.

Several researchers have investigated the performance of ejector refrigeration systems, both experimentally and numerically [3-7]. Predicting performance remains a critical part of studies on ejector refrigeration systems. In a widely used study, Huang et al. [8] considered the performance of an
ejector system using R141b as a refrigerant. Both experimental and numerical results were presented with significant errors in the numerical results since ejector loss coefficients were taken to be independent of generator and evaporator pressures. Chen et al. [9] theoretically investigated the performance of an ejector refrigeration system under overall working modes. Ejector coefficients were matched with the already available results and significant errors were shown to exist. To address the issues inherent in the modelling ejector performance, Li et al. [10] recently suggested a way of determining ejector loss coefficients using the sparsity-enhanced optimization methods. Reduced prediction errors were obtained; however, the method is complicated and time consuming.

As the reviewed studies and the literature indicate, most studies on ejector performance have considered working fluids that are not environmentally benign. Most of the considered refrigerants possess high ozone depletion potentials (ODPs) and/or high global warming potentials (GWPs). With the increasing concerns of climate change, all ozone depleting refrigerants have been banned and there is increasing regulatory pressure to eradicate the use of refrigerants with high GWPs [11,12]. With this, there are emerging refrigerants with zero ODP and with low global warming potentials that are being implemented in refrigeration systems. Among these, hydrofluoroolefins are potential replacements for the conventional harmful working fluids in refrigeration systems. A few studies have shown the potential of hydrofluoroolefins in ejector refrigeration systems [13–15]. However, the performance of these refrigerants in ejector refrigeration systems is not yet fully established. There is still a lack of performance data and curves to size ejector refrigeration systems working with these refrigerants.

The main aim of this study is to characterize the performance of an ejector refrigeration system working with HFO-1233zd(E), a new environmentally benign working fluid. Moreover, as an extension of the authors’ previous study [13], correlations for the optimal COP and optimal generator temperatures that can be used in the design and selection of these systems have been derived and presented. To the authors’ knowledge, no studies have optimized ejector refrigeration systems working with R1233zd(E).

2. Ejector-based air conditioning system

Figure 1 shows an air conditioning system working with an ejector refrigeration system. In this case, the ejector sub-system replaces the conventional refrigeration system and eliminates the compressor.

![Figure 1. Ejector based air-conditioning system for space heating and cooling](image)

The pump used to raise the condensate pressure to the generator pressure uses less energy compared to a compressor in a vapor compression system. The cooling effect from the refrigerator either cools down the water or the air that later circulates in the conditioned space. The ejector system is thermally activated, giving flexibility on the type of fuel that can be used. In this case, solar thermal is used to...
drive the generator with an auxiliary heater included in cases where the solar resource is not available or insufficient. There is also a possibility of incorporating a ground heat exchanger for improved performance, especially in locations where the ambient temperatures are extremely hot or cold. With a ground loop, the system utilizes the nearly constant ground temperatures and gives much better coefficients of performance.

The ejector is a central component of the entire system; its design and size will significantly affect performance. In the ejector, a high pressure vapor produced in the generator expands through the nozzle. Inside the nozzle, the reduction in the primary flow pressure leads to a higher velocity, creating a vacuum that entrains the secondary fluid from the evaporator creating a refrigeration effect. During the cooling season, water is cooled as it passes through the evaporator and is circulated through the conditioned space to achieve the required cooling. During the heating season, the heat rejected by the condenser is used to raise the water temperature which is then circulated to the heated space.

As the ejector sub-system is the central component of the system, this study focuses on the detailed performance investigation of the ejector using an environmentally benign working fluid which was shown to have better performance than R141b and R245fa [13]; the two refrigerants that have been widely studied in ejector refrigeration systems.

3. Theoretical analysis

A detailed and more precise model that determines the performance of an ejector refrigeration system using ejector loss coefficients that are dependent on the generator temperature and the evaporator temperature as well as the ejector area ratio was developed previously [13]. This model has been adopted in this study. The performance of an ejector can be characterized by the entrainment ratio, which is the ratio of the secondary mass flow rate to the primary mass flow rate as

\[ \mu = \frac{\dot{m}_s}{\dot{m}_p} \]  

The primary mass flow rate, \( \dot{m}_p \), is given by

\[ \dot{m}_p = \frac{P_g A_t}{\sqrt{T_g}} \sqrt{k \left( \frac{2}{R} \left( \frac{k + 1}{k - 1} \right) \right) \sqrt{\eta_p}} \]  

where \( T_g \) is the generator temperature, \( P_g \) is the generator pressure corresponding to the saturation pressure of the generator temperature, \( R \) represents the specific gas constant, \( A_t \) is the nozzle throat diameter, \( k \) is the heat capacity ratio and \( \eta_p \) is the nozzle efficiency.

The secondary mass flow rate is given by

\[ \dot{m}_s = \frac{P_e A_{2s}}{\sqrt{T_e}} \sqrt{k \left( \frac{2}{R} \left( \frac{k + 1}{k - 1} \right) \right) \sqrt{\eta_s}} \]  

in which \( \eta_s \) is the isentropic efficiency, \( P_e \) is the evaporator pressure, \( T_e \) is the evaporator temperature and \( A_{2s} \) gives the area occupied by the secondary flow at the mixing plane. The determination of this area requires knowledge of the area occupied by the primary flow at mixing, \( A_{2p} \). The equations used for this purpose are available in Mwesigye and Dworkin [13].

Another important parameter useful in classifying the performance of ejector refrigeration systems is the coefficient of performance (COP) given as

\[ \text{COP} = \frac{\dot{Q}_e}{\dot{W}_e} \]  

After simplification and using the energy rate balances for the energy transfer by heat from the evaporator and the generator and the work needed by the pump, the COP becomes

\[ \text{COP} = \mu \left( \frac{h_{eo} - h_{ei}}{h_{go} - h_{gi}} \right) \]  

where \( h_{eo} \) and \( h_{ei} \) are the enthalpies at the exit and the inlet of the evaporator, \( h_{go} \) and \( h_{gi} \) are the enthalpies at the exit and the inlet of the generator and \( h_{eo} \) is the enthalpy at the condenser exit.
4. Solution methodology

This work uses an improved ejector model developed in Mwesigye and Dworkin [13] where the ejector loss coefficients are determined as functions of the ejector area ratio and evaporator and generator pressures. The detailed model, equations used and loss coefficients are available in [13]. The performance is investigated with ejectors having area ratios between 6.44 and 10.98 as shown in Table 1. These ejector area ratios have been selected based on related studies in the literature for the considered range of generator temperatures [8]. The other parameters considered in this study were: the generator temperature, which was taken to be between 70°C and 110°C, the evaporator temperature between 0°C and 16°C and the condensing temperature between 20°C and 45°C. The generator temperatures are selected to be lower than the critical temperatures of the used refrigerants and to give reasonable ejector performance since the COP reduces with increasing generator temperatures. The evaporator temperature is related to the required cooling effect, whereas the condensing temperature is inline with ambient temperatures commonly encountered. Equations (1) to (4) and the detailed model equations in Mwesigye and Dworkin [13] are solved iteratively using Engineering Equations Solver (EES) using a program written for this purpose. For the range of temperatures in this study, R1233zd(E) is nearly an isentropic working fluid, eliminating the need for working fluid superheating and ensuring expansion in the ejector with less friction losses since no liquid droplets will be present.

Table 1. Ejector geometries considered in this study

| Ejector | $d_1$ (mm) | $d_2$ (mm) | $d_3$ (mm) | $A_r$ [-] |
|---------|------------|------------|------------|-----------|
| AA      | 2.64       | 4.5        | 6.70       | 6.44      |
| AC      | 2.64       | 4.5        | 7.60       | 8.28      |
| AD      | 2.64       | 4.5        | 8.10       | 9.41      |
| AG      | 2.64       | 4.5        | 7.34       | 7.73      |
| EH      | 2.82       | 5.1        | 9.20       | 10.64     |

5. Results and discussion

An ejector refrigeration system operates in two modes; the critical mode and the subcritical mode. The critical mode, which gives the highest possible performance is achieved when the condensing pressure is lower than the critical pressure. Below the critical pressure, the coefficient of performance and the entrainment ratios stay constant. In this mode, both the primary flow and the secondary flow are choked. As the pressure increases above the critical pressure, the performance of the ejector declines and only the primary flow is choked. As the ejector condensing pressure increases further, eventually a point is reached at which no more secondary flow is entrained, leading to ejector failure. This phenomenon is demonstrated in Fig. 2(a) and (b). As the figures show, small area ratios give an ejector that works in both the critical and subcritical modes. As the area ratios increase, it becomes difficult to keep both the flows in the choked mode. As such, significantly higher generator temperatures and therefore generator pressures are required to keep the ejector in the critical mode. As Fig. 2(b) shows, higher COPs are attainable, but the generator temperatures should increase for the ejector to work in the critical mode of operation. Also shown is that lower generator temperatures give better performance, but they are associated with much lower temperatures at which the ejector ceases to work. At a fixed evaporator temperature, increasing the generator temperature will give higher mass flow rates of the primary flow and therefore reduce the entrainment ratio even when the secondary fluid flow rate has not changed.

Figure 3 shows the values of the COP at the critical point as a function of generator temperature. At a given generator temperature Fig. 2(a) and (b) showed the COP to stay constant under the critical mode of operation as the condensing temperature changes. In the critical mode of operation, the performance reduces as the generator temperatures rise and as evaporator temperatures decrease as shown in Fig. 3(a) and (b). At a given generator temperature, lower evaporator temperatures make the entrainment of the secondary flow difficult leading to a lower refrigeration effect in the evaporator. Whereas higher generator temperatures at a fixed evaporator temperature mean higher primary flow rates and reduced area in the mixing section for the entrainment of the secondary flow. In Fig. 3, higher performance is
shown as the area ratio increases from 6.44 to 10.64, however, as discussed earlier, the ejector breakdown temperatures are lower as the area ratios increase. Therefore, operation at higher generator temperatures is recommended to maintain performance in the critical mode. This depends on the area ratio considered, but generally generator temperatures above 90°C ensure the ejector works in the critical mode for condensing temperatures lower than 30°C for all area ratios considered and evaporator temperatures above 12°C.

![Figure 2](image_url)

**Figure 2.** COP as a function of condensing temperature at different evaporator temperatures with $T_e = 12^\circ C$ (a) $A_r = 6.44$ and, (b) $A_r = 10.64$

![Figure 3](image_url)

**Figure 3.** Critical COP as a function of generator temperature at different evaporator temperatures (a) $A_r = 6.44$, (b) $A_r = 10.64$. The actual mode of operation can be determined keeping the condensing temperature constant and varying the generator temperature at given area ratios and evaporator temperatures. Results show that the COP increases with generator temperature, attains a maximum, and reduces again with a further increase of the generator temperature. This analysis is done by increasing the generator temperature and obtaining the actual performance of the ejector, not just the critical COP values. Both COP values in the subcritical and the critical mode of operation are recorded for the ejector depending on the input parameters and plotted as shown in Fig. 4. The figure indicates that increasing the area ratio requires higher generator temperatures for optimal conditions to be achieved. While lower area ratios give lower optimal generator temperatures, the optimal generator temperature slightly reduces as the evaporator temperature increases as shown in Fig. 4(b).

![Figure 4](image_url)

**Figure 4.** COP as a function of generator temperature at a condensing temperature of 35°C and (a) different area ratios for $T_e = 8^\circ C$, and (b) different evaporator temperatures for $A_r = 8.24$. 
Correlations for the optimal generator temperature and the optimal COP that can be used in the preliminary design and sizing of ejector refrigeration systems working with R1233zd(E) were obtained. The optimal generator temperature is given by

\[ T_{g,\text{opt}} = 20.887 A_r^{0.204} \left( \frac{T_e}{T_e} \right)^{1.4223} \left( \frac{T_c}{T_c} \right)^{0.3804} \]  

(5)

Whereas the optimal COP is determined according to

\[ \text{COP}_{\text{opt}} = 0.5303 A_r^{0.4338} \left( \frac{T_e}{T_e} \right)^{18.825} \left( \frac{T_c}{T_c} \right)^{-18.519} \]  

(6)

where \( T_e / 273.15 \) and \( T_c / 273.15 \) (with \( T_e \) and \( T_c \) in K). The correlations are valid for \( 6.44 < A_r < 10.64, 30 < T_c < 40^\circ C, \) and \( 4 < T_e < 16^\circ C \) giving generator temperatures between 70\(^\circ C\) and 110\(^\circ C\).

6. Conclusion

In this study, the performance of an ejector refrigeration system using an environmentally benign working fluid, R1233zd(E) has been investigated. A novel model that determines performance with improved accuracy incorporating ejector loss coefficients as functions of the area ratio and the ejector pressure lift was used. The operation in both the critical and subcritical modes of operation has been established and the optimal values of the generator temperature and coefficient of performance have been determined. Generator temperatures higher than 90\(^\circ C\) are shown to keep the ejector in the critical mode of operation for condensing temperatures lower than 30\(^\circ C\) and evaporator temperatures greater than 12\(^\circ C\). The optimum COPs in the range 0.068 and 0.88 were obtained for the range of parameters considered in this study. Correlations that can be used to determine the optimal generator temperature and the optimal coefficient of performance at given ejector area ratios and required evaporator temperatures for an ejector working in a known environment have been derived and presented for the first time.

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