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EXPERIMENTAL STUDY OF AN EJECTOR REFRIGERATION SYSTEM

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ABSTRACT

The aim of the present work is to experimentally study the possibility of improving the energy efficiency of a vapour compression refrigeration system where a two-phase ejector replaces the expansion valve. A test bench using refrigerant R134a was designed and built which functions in both the conventional mode and in ejector mode. The primary nozzle of the ejector was equipped with a double throat, having an adjustable area for the first throat and a fixed area for the second throat. Experimental results showed an improvement of 11% in the coefficient of performance (COP) in ejector mode as compared with the conventional mode. The role of the double throat in the primary nozzle as well as the behaviour of the pressure ratio and entrainment ratio parameters are discussed. A modified ejector refrigeration system using two evaporators is proposed as a means of improving the control stability and addressing the separator effectiveness limitations.

1. INTRODUCTION

The first ejector was invented by Charles Parson around 1901 and was used to remove air from steam engine condensers. The first use of an ejector in a refrigeration application was by Maurice Leblanc in 1910, again using steam as the motive force in the primary of the ejector (Chunnanond and Aphornratana, 2004). More recently interest has grown in using ejector technology to increase the efficiency of modern vapour compression refrigeration systems. Menegay and Kornhauser (1996) showed in a theoretical study using R12 that an improvement of 21% in the COP of a vapour compression system was possible where a two-phase ejector replaces the traditional thermostatic expansion valve. Nakagawa and Takeuchi (1998), using R134a, reported an estimated 10% COP improvement based on their experimental results. Takeuchi *et al*. (2004) described the use of a two-stage primary nozzle as part of their method of optimizing the two-phase ejector refrigeration cycle. Chaiwongsa and Wongwises (2008) completed experimental studies with R-134a using a two-phase ejector as an expansion device. In their setup the evaporator was flooded and the ejector serves to partly recirculate the refrigerant on the low pressure side of the system. As an engineering master's thesis project an experimental test bench was designed and built with the aim of verifying and improving the COP of a refrigeration system where a two-phase ejector replaces the thermal expansion valve.

2. EXPERIMENTAL SETUP

2.1 The two-phase ejector

Figure 1 presents a simplified schematic of the ejector used in the test bench. The high pressure liquid refrigerant arrives from the condenser and enters the primary nozzle. This flow stream is called "the primary". The low pressure gaseous refrigerant arriving from the evaporator is called "the secondary". The exiting of the primary stream at high speed from the primary nozzle creates a low pressure, inducing the secondary stream to enter the ejector and accelerate in the central portion of the ejector. As the two flow streams arrive at the end of the mixing section they have almost become a homogenous flow. The pressure increase in the diffuser portion of the ejector will reduce the work required by the compressor and thus improve the COP.

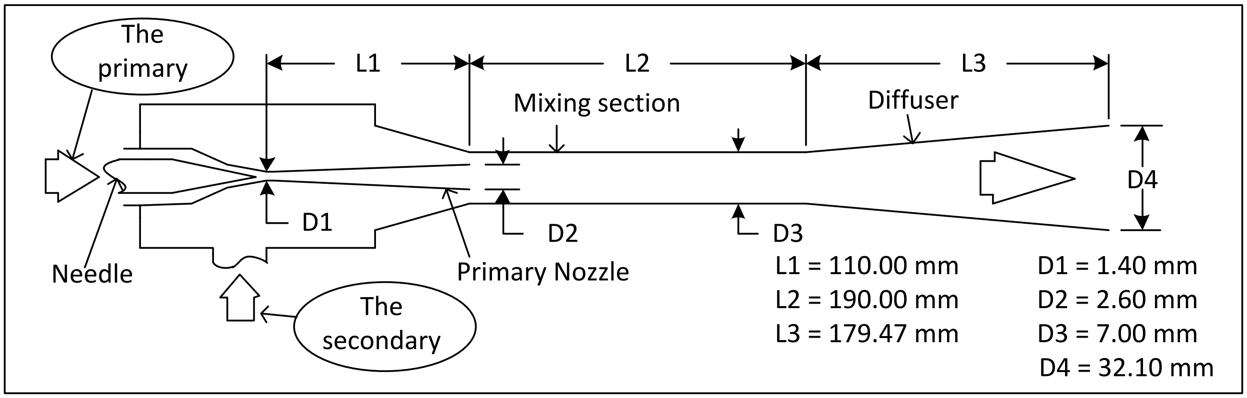

Figure 1: Schematic of ejector mode configuration

2.2 Ejector mode and conventional mode

The experimental test bench was designed to function in two possible modes. The nominal refrigeration capacity of the evaporator was 5kW. The compressor was a Carlyle variable speed semi hermetic piston model 06DR013CC150. The compressor frequency was maintained at 35 Hz, where 60 Hz corresponds to the top speed of 1750 RPM. Figure 2 shows a simplified view of the flow arrangement in ejector mode. In conventional mode the ejector and separator were isolated from the circuit and the traditional thermostatic expansion valve was used. In both configurations two auxiliary circuits were used, identified as the “SOURCE” and “SINK”. The SOURCE circuit provided a constant flow rate of a 40% volume ethylene glycol water solution to the evaporator at a controlled temperature of 1.1°C. For all experimental runs the refrigerant evaporator entrance temperature target was -5°C. The SINK circuit provided a constant flow rate of water to the condenser such that the exiting refrigerant temperature remained at 40°C. The test bench was equipped with 14 RTD temperature probes, 5 pressure sensors and three flow meters. A data acquisition system allowed the measurement and recording of all of the experimental data.

Figure 3 presents a more detailed schematic drawing of the test bench, showing the placement of the instrumentation with the identification of the thermodynamic states at the points of interest. The configuration shown is for the conventional mode.

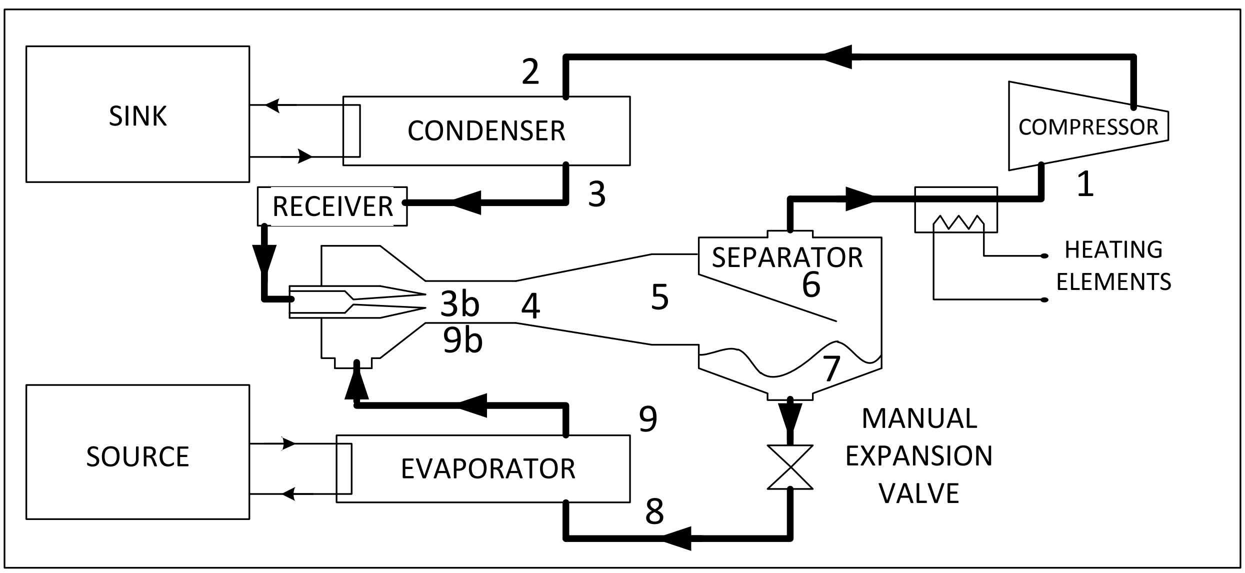

Figure 2: Schematic of ejector mode configuration

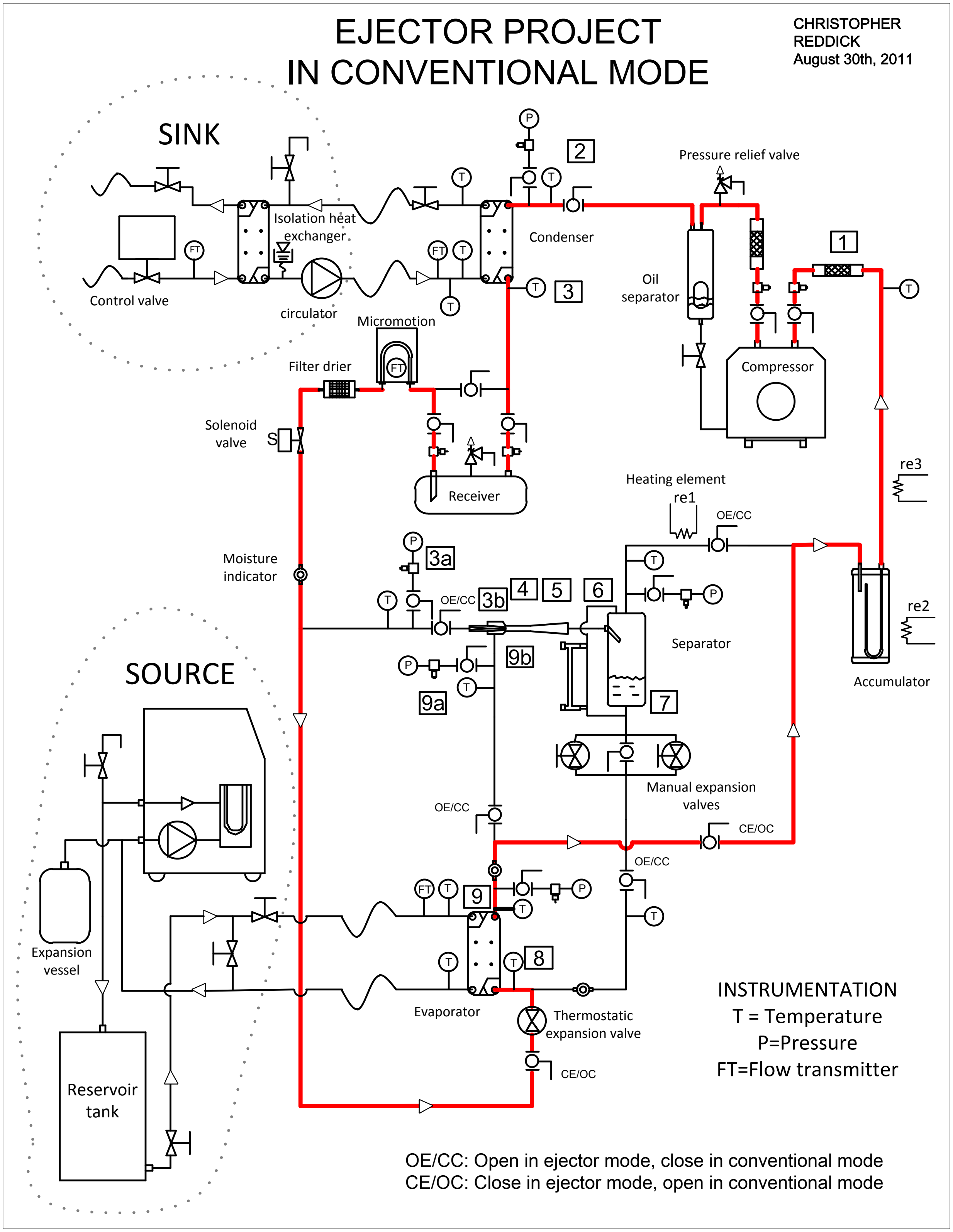

Figure 3: Schematic of conventional mode configuration

3. MEASUREMENT RESULTS AND DISCUSSION

During the preliminary runs and calibration period in ejector mode it was determined that the separator was not effectively separating the liquid phase from the gas phase. In order to benefit from the unused cooling effect returning to the compressor, three electric heating elements were installed between the separator exit and the compressor entrance. For the purpose of effectively comparing the COP of the two operating modes, all of the experimental runs in section 3.1 were completed with a target of 5°C for the refrigerant entering the compressor. In conventional mode this was achieved by adjusting the superheat screw on the thermostatic expansion valve. In ejector mode the electric heaters were controlled using rheostats to achieve the 5°C target. The COP was calculated using equation (1), where Q_EV, Q_re and Q_CM refer respectively to the thermal power exchanged in the evaporator, the heating elements and the compressor. In conventional mode the Q_re term is zero.

$$COP = (Q\_EV + Q\_re)/Q\_CM \qquad (1)$$

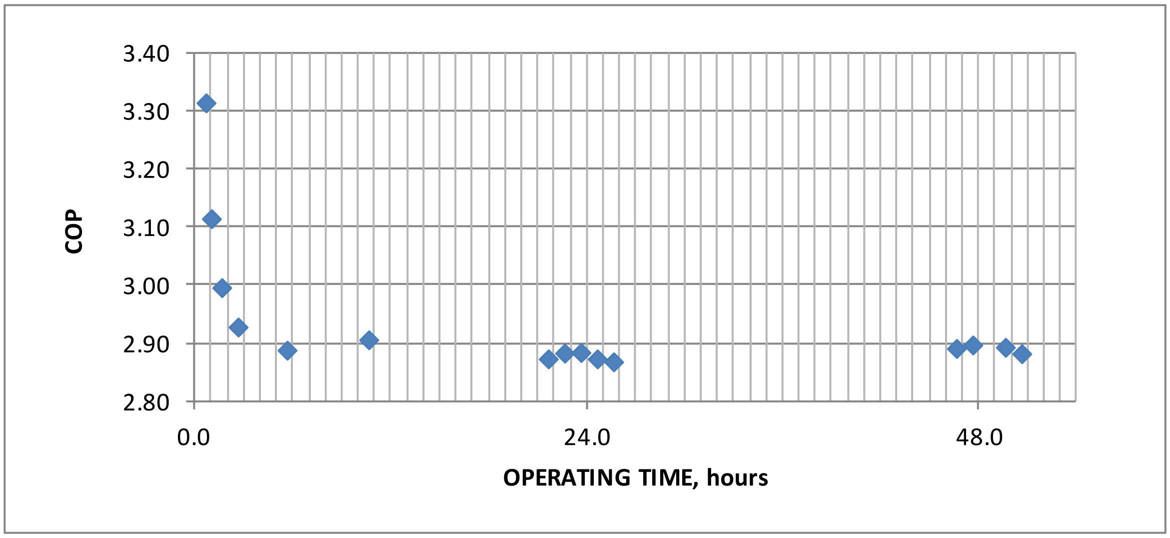

Figure 4: COP in conventional mode

3.1 COP: Ejector mode versus conventional mode

During a 51 hour period 15 runs in conventional mode and 12 runs in ejector mode were carried out. Each run consisted of 21 sets of recorded data, where each data set was collected every 30 seconds for 10 minutes. The test bench was started in conventional mode. As shown in Figure 4, after 5 hours of operation in conventional mode the test bench had reached steady state conditions. The reason for this behaviour in the COP is the fact that while the compressor has not reached its steady state temperature, the temperature at the compressor exit is lower, leading to a smaller estimate of the compressor power consumption and thus an artificially high COP value.

While in ejector mode, 3 runs were carried out for each of the 4 chosen primary needle valve opening positions of 0.30mm, 0.36mm, 0.38mm and 0.41mm. As seen in Figure 5, in ejector mode the COP is a function of both compressor entrance refrigerant temperature and the needle opening. For a given needle opening the COP is a linear function of compressor entrance refrigerant temperature. The average COP of the three ejector mode runs having a needle opening of 0.38mm is 3.19, while the average COP of the three conventional mode cases having the closest compressor entrance refrigerant temperature is 2.88. Thus an improvement of 11% is found in the COP in ejector mode as compared to conventional mode.

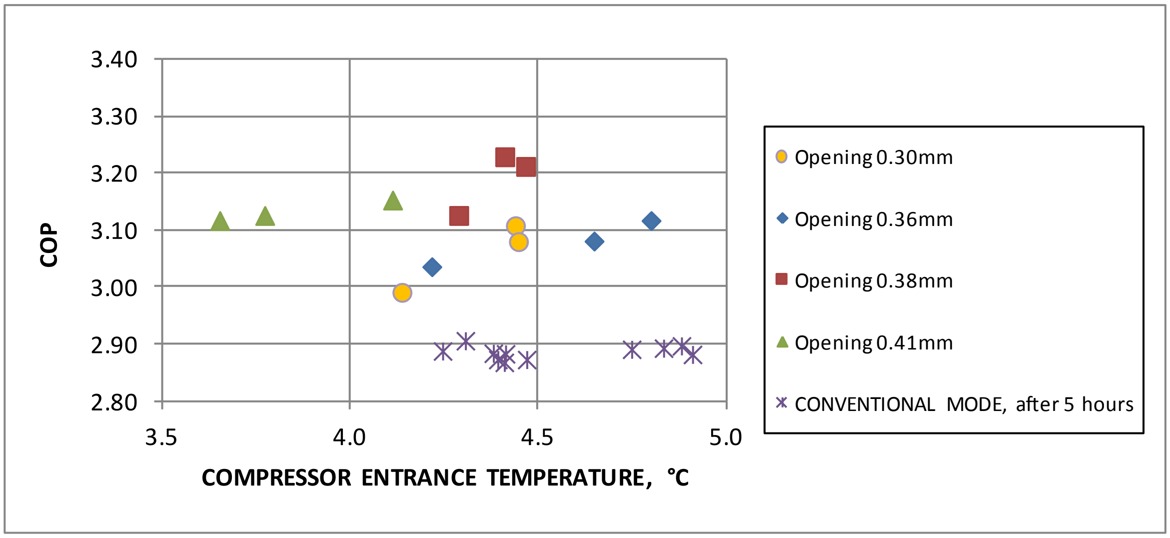

Figure 5: COP in conventional mode and ejector mode

3.2 Pressure Enthalpy Graph in ejector mode

The thermodynamic states of each of the points of interest in the cycle are identified in both Figures 2 and 3. The calculated thermodynamic states of one of the ejector runs having a needle opening of 0.41mm are shown in Figure 6. The relative positions on the graph are slightly exaggerated in order to better visualize the cycle. The internal states of the ejector are assumed to be at the saturation pressure corresponding to -8°C, being 217.4 kPa, for the purpose of calculating and visualizing the points 3b, at the primary nozzle exit, 9b, at the secondary exit, and 4, at the end of the mixing section.

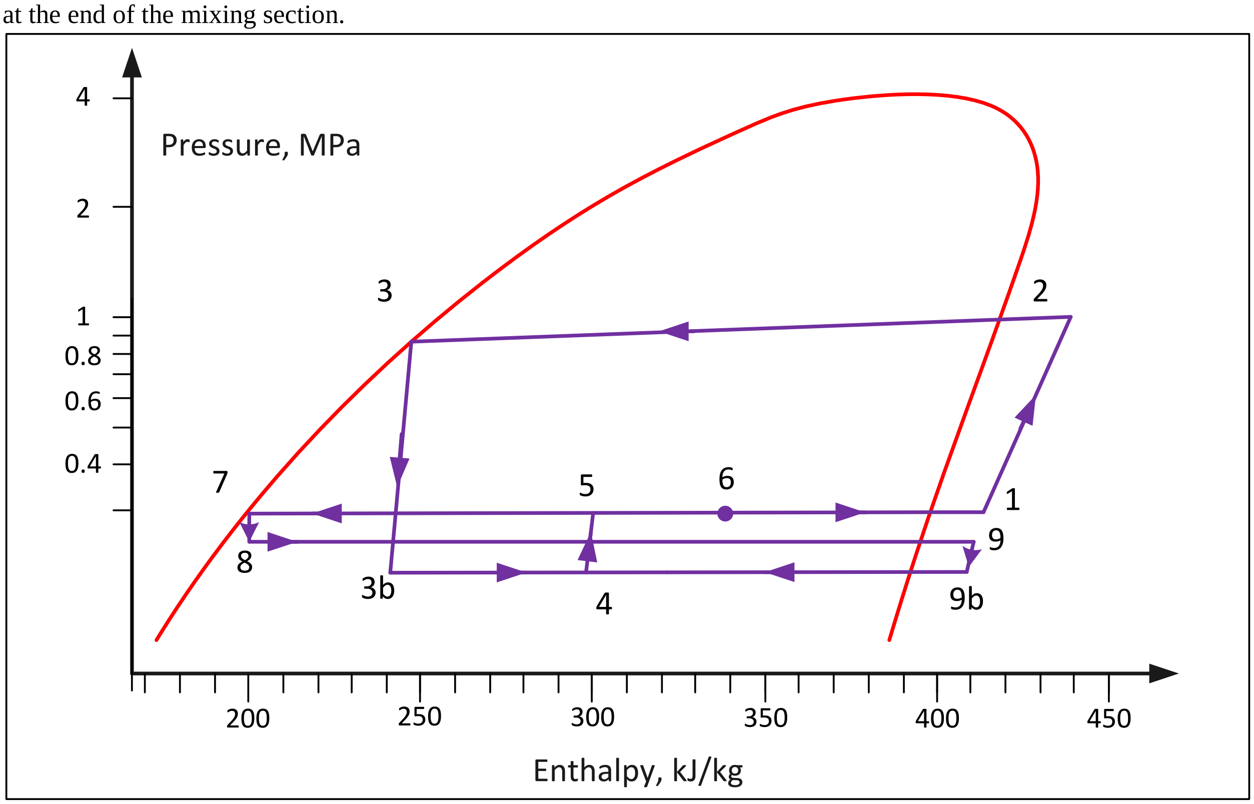

Figure 6: Pressure Enthalpy graph in ejector mode

Superheated refrigerant enters the compressor at 1 and leaves at 2. At the condenser exit 3 the refrigerant is assumed saturated as a pressure reading is not available. Passing through the primary nozzle it is assumed that that refrigerant flows isentropically to 3b. The exit plane of the primary nozzle is also the point at which the secondary 9b enters. These two streams mix in the central part of the ejector ending at point 4. Passing through the ejector diffuser the pressure increases to point 5. The refrigerant vapour in the separator at point 6 has a quality of 0.71 and thus the vapour flow leaving the separator clearly contains a significant amount of liquid. The electric heating elements increase the enthalpy to state 1. At the lower exit from the separator the liquid stream 7 passes through a manual expansion valve and enters the evaporator at state 8. The superheated vapour leaving the evaporator at 9 then passes into the secondary and is slightly accelerated to state 9b and combines with the primary stream. The very low vapour content of the refrigerant entering the evaporator at 8, being around 1.4%, contributes to the improvement of the COP in ejector mode. As a comparison with the conventional mode, the vapour content of the refrigerant entering the evaporator is 31%.

3.3 Needle opening and the double throat

The maximum cooling effect of the evaporator occurred when the primary needle was opened to 0.35mm, as shown in Figure 7. This curve has essentially the same shape as the secondary mass flow as a function of needle opening. At this needle opening the area of the fixed throat, of 1.54 mm^2, is equal to the area of the variable throat. This suggests that both the presence of a double throat and the form of the throat play an important role in the formation of small bubbles that serve as nucleation sites in the two-phase behaviour of the refrigerant. When the throats have the same area, the gap between the needle and the nozzle is 0.12mm for the variable throat, compared to a diameter of 1.4mm at the fixed throat. Figure 8 shows further details of the primary nozzle.

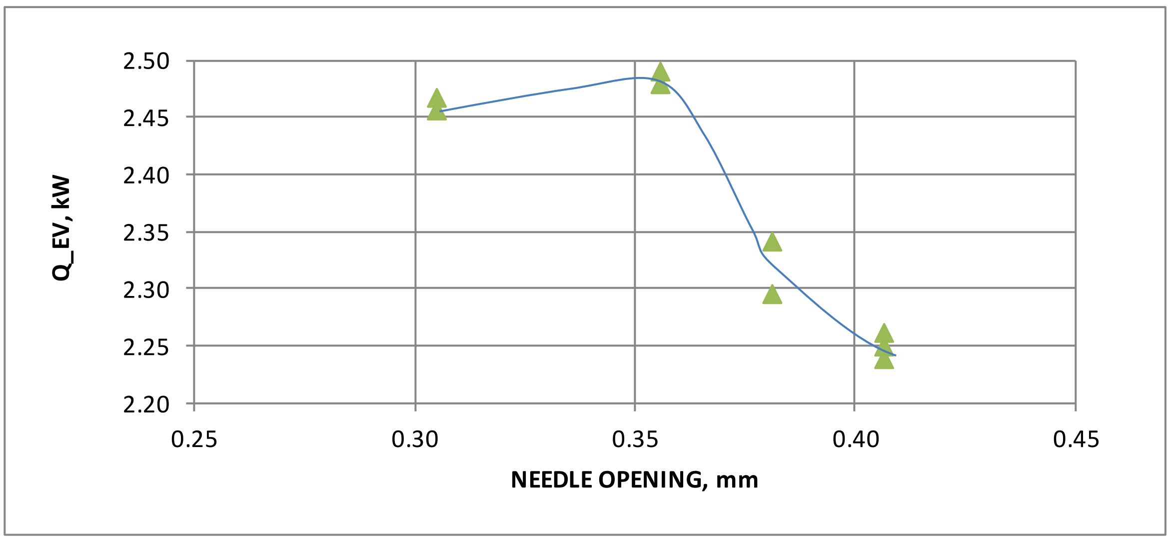

Figure 7: Evaporator cooling as a function of needle opening

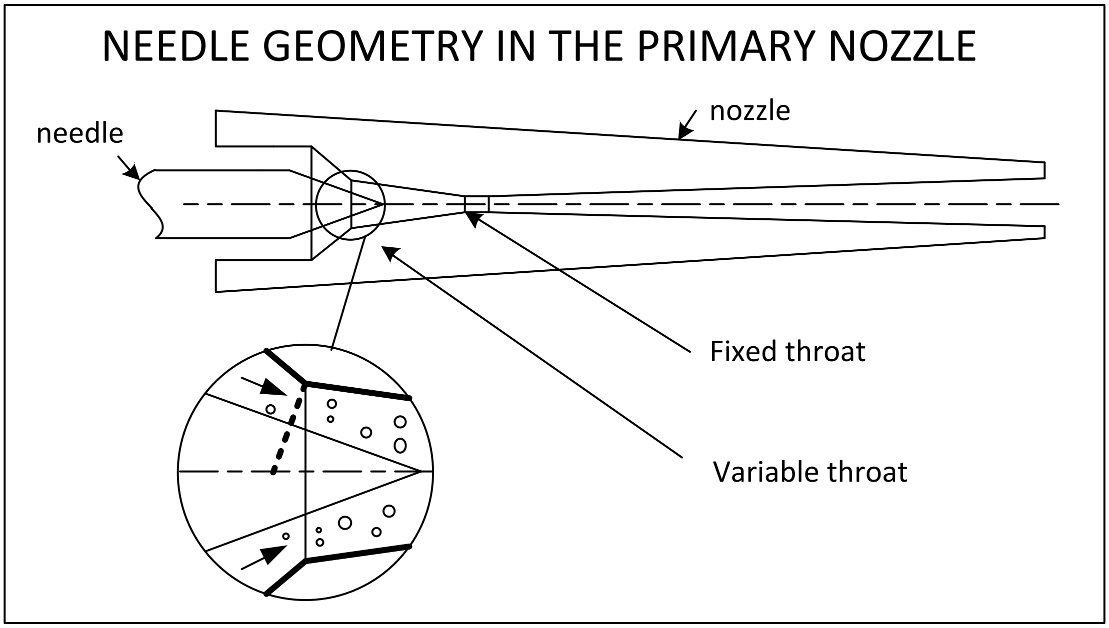

Figure 8: Primary nozzle details

3.4 Ejector pressure ratio and entrainment ratio

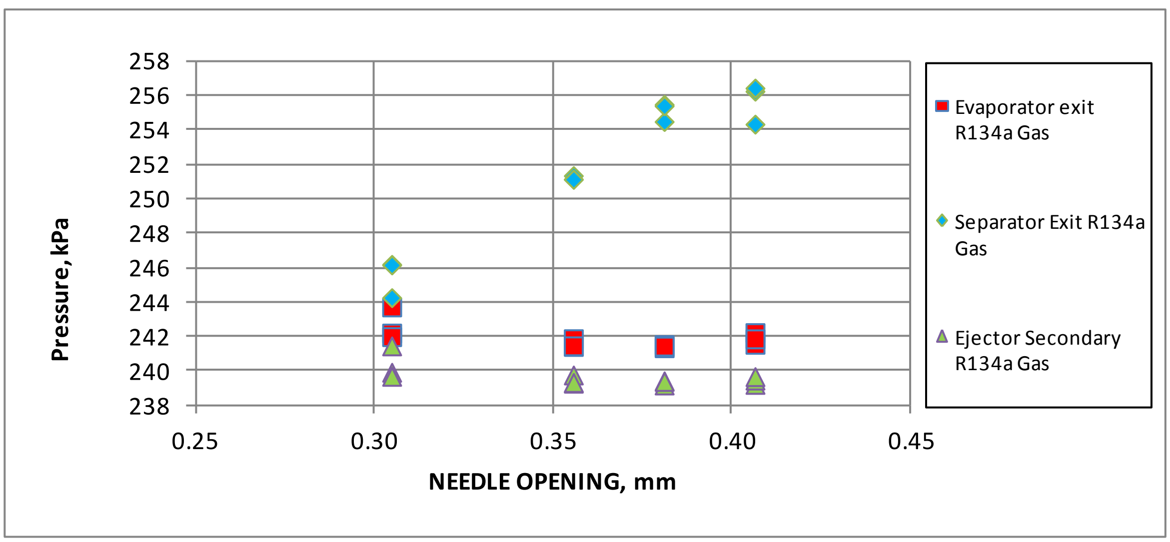

Figure 9: Ejector pressure trend

As shown in Figure 9, the measured pressure increase between the secondary entrance and the separator was very modest, being around 17 kPa. The form of the separator gas pressure curve closely resembles the form of the primary mass flow rate versus needle opening curve.

The pressure ratio "r" and entrainment ratio "ω" are often used to characterize ejectors. The pressure ratio r is defined as the ratio of the pressure at the ejector exit, being the separator pressure, to the pressure of the secondary stream entrance. As shown in Figure 10, the pressure ratio r increases and levels off as the needle is opened. The leveling off of the pressure ratio as the needle is opened more that the 0.38mm opening indicates that the primary throat controls the flow rate at this point and that opening the needle beyond this point does not increase the primary flow rate. It is possible that a further increase in pressure ratio might be possible by increasing the compressor

RPM. The current test bench will not allow this because the installed heating elements do not have sufficient kW capacity.

The entrainment ratio ω is defined as the ratio of the secondary mass flow rate to the primary mass flow rate. As shown in Figure 11, ω decreases as the needle is opened. The entrainment ratio is relatively constant for the first two set points of 0.30mm and 0.36mm but dropped significantly for the 0.38 and 0.41 openings. This suggests that the loss of the double throat effect for the needle openings of 0.38mm and 0.41mm contributed to the decrease in the entrainment ratio.

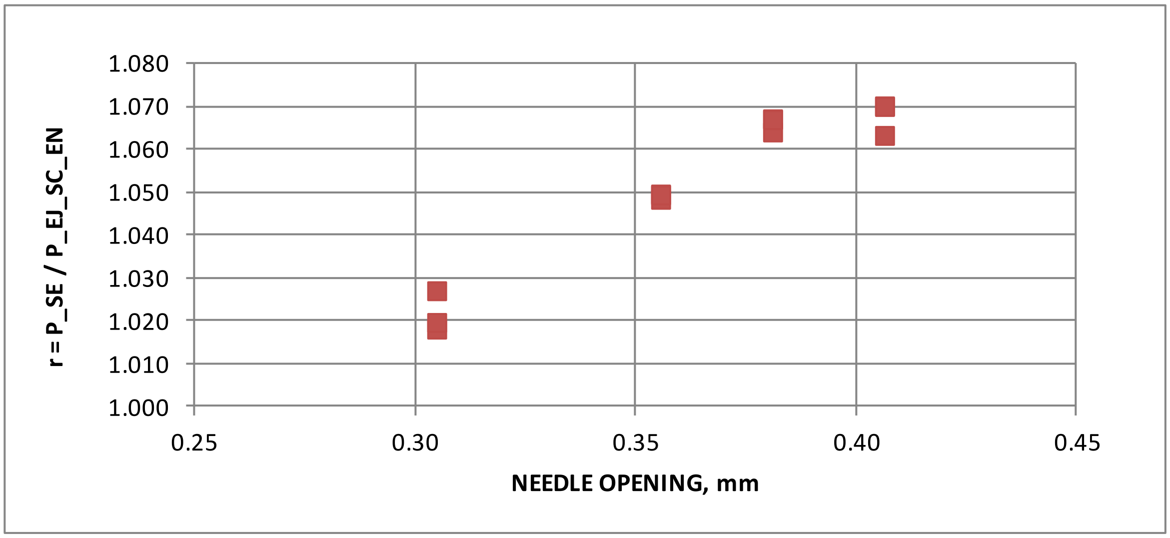

Figure 10: Pressure ratio r

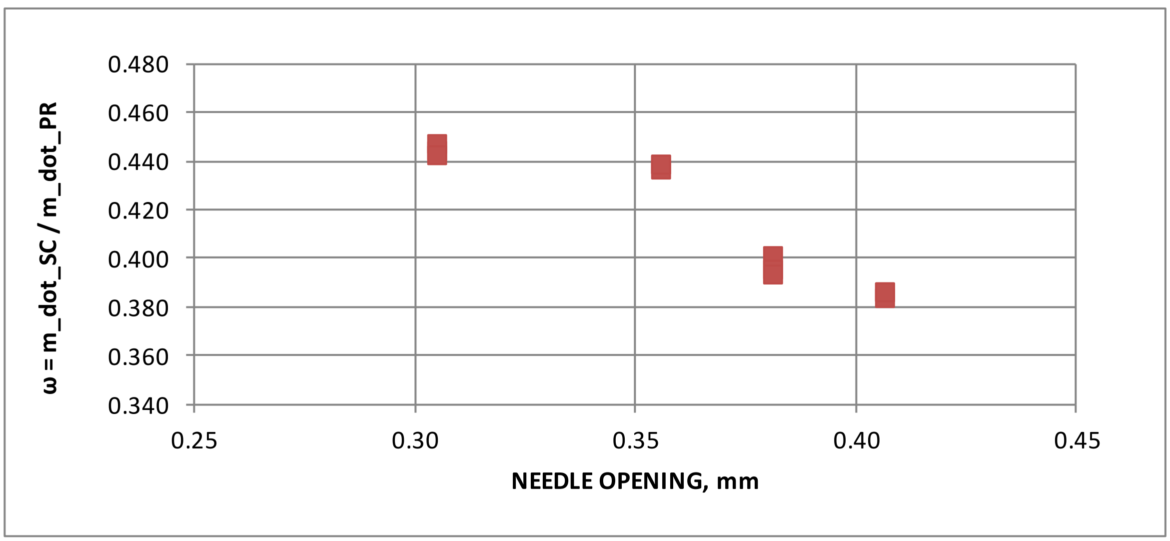

Figure 11: Entrainment ratio ω

3.4 Proposal for improvement

During the experimental procedure it was apparent the next phase of development work in two-phase ejector vapour compression must have as a priority the improvement of the control aspect of the studied system. The use of the ejector creates the need to manage two cooling effects. The primary evaporator plays its traditional role. The need to control the superheat of the flow leaving the separator suggests the use of a secondary evaporator such as proposed in Figure 12.

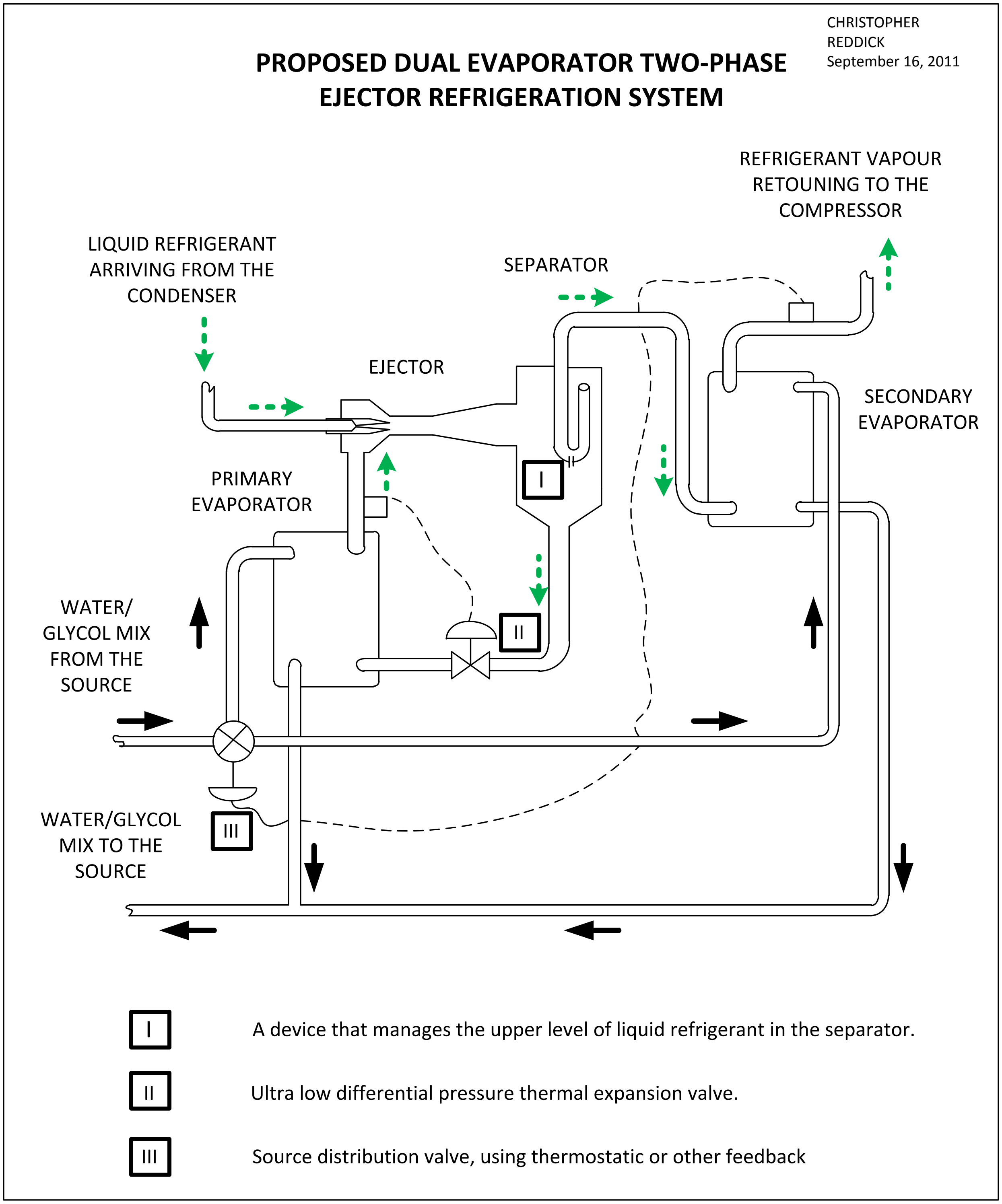

Figure 12: Dual evaporator two-phase ejector refrigeration system

4. CONCLUSIONS

1. An improvement of 11% was found in the COP of a vapour compression refrigeration system where the expansion valve is replaced by a two-phase ejector using R134a. This comparison was based on an experimental test bench operating in both ejector and convention vapour compression modes.
2. Future development work on two-phase ejectors must place a priority on solving the problem of controlling the amount of superheat at the separator exit.
3. The double throat effect plays an important role in the creation of nucleation sites in the primary nozzle.

NOMENCLATURE

COP	coefficient of performance	
m_dot_PR	mass flow rate of the primary	(g/s)
m_dot_SC	mass flow rate of the secondary	(g/s)
P_SE	pressure in the separator	(kPa)
P_EJ_SC_EN	pressure in the ejector secondary entrance	(kPa)
Q_CM	thermal power in the compressor	(kW)
Q_EV	thermal power in the evaporator	(kW)
Q_re	thermal power in the heating elements	(kW)
r	ejector pressure ratio (P_SE/P_EJ_SC_EN)	
RTD	resistance temperature detector	
ω	entrainment ratio (m_dot_SC/m_dot_PR)	

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