Solar Cooling Using Variable Geometry Ejectors

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ABSTRACT

Solar heat driven cooling systems are an attractive concept. The need for cooling is associated with high ambient temperatures. Ejector based cooling systems have been in existence for some time but have not gained widespread use due to their low performance and difficulty of control. Furthermore, although ejectors are well suited to steady state operation, they do not couple well to varying solar conditions. However, ejectors offer a robust and reliable design which demonstrates flexibility to a range of refrigerants and most importantly, a low electrical power requirement.
Variable geometry ejectors help to overcome the present objections of ejectors by taking advantage of cooler ambient conditions and allowing continued operation at elevated ambient temperatures. This paper describes and compares performance maps generated by fixed and variable geometry ejectors and hints at control strategies for each.

INTRODUCTION

The ejector principle has been known for over 100 years, originally developed for evacuating air from condensors of steam engines. Low pressure steam was extracted from a boiler to power the ejector which then produced a moderate vacuum. It was a natural progression for steam based ejectors to be used to draw vapour from an evaporator and thus produce cooling effect. Ejectors were typically applied to cooling applications when there was a ready source of low pressure steam and were commonplace in ships and hotels from around 1910 to 1930. They typically provided cooling with no maintenance over extended periods which made them popular.

The development of chlorinated fluorocarbon refrigerants in the 1930s allowed electrically driven heat pumps to be deployed to locations such as houses where a steady source of heat may not have been available but electricity was. The superior thermal performance and safe operation of heat pumps were other notable benefits.

Heat pump cooling systems dominate the air-conditioning market to the present day. However, concerns over electricity network peak loading, ozone depletion and high greenhouse gas emissions associated with electrically driven cooling have led researchers to consider solar heat driven cooling systems, most prominently since the 1990s.

The Ejector Heat Pump

The ejector is a thermally driven compressor. In a heat pump system, the ejector takes the place of the electrically driven compressor, but uses heat rather than electricity to produce the compression effect. (figure 1). The ejector has no moving parts and is simple and reliable which make it attractive for commercial production. However, the thermal efficiency of the ejector is low which implies that the ejector requires a large solar collector and large condenser to operate in a heat pump application. Thus the savings in electricity consumption must be compared with the additional cost of the solar collector.

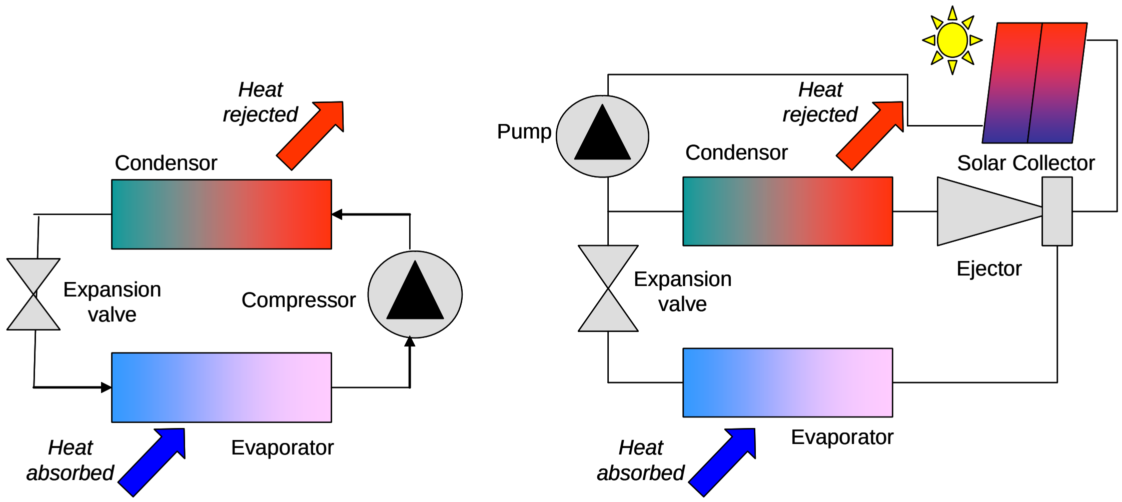

Fig.1: An ejector circuit compared to a conventional heat pump

A pump is required to generate a pressure difference for the heat pump to operate, but since liquid is being compresses, the electricity required is relatively small. All other components in the heat pump circuit are conventional.

The Ejector Principle

An ejector consists of two converging-diverging nozzles (figure 2). The primary nozzle has a small throat diameter and produces a supersonic jet from the generator flow. Since the flow has accelerated from essentially stagnation conditions at the nozzle entrance to such high velocity, its pressure and temperature must drop due to conservation of energy (figure 3). Thus the prime function of the primary jet is to create the suction equivalent of the electrical compressor. Oblique shock waves are produced at the primary nozzle exit as the flow enters a higher pressure mixing chamber. Thus the primary jet operation is a compromise between low pressure and reduced ejector efficiency due to the entropic shock waves.

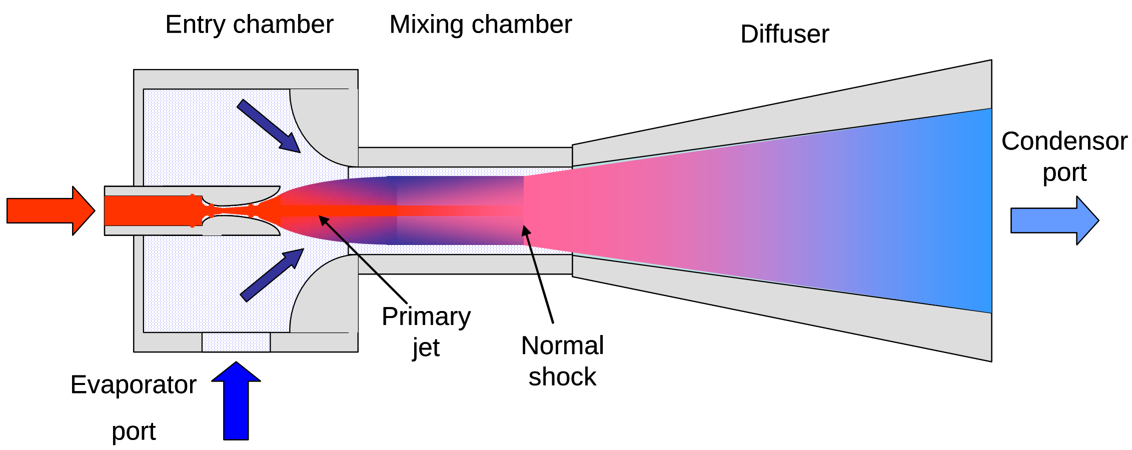

Fig. 2: Ejector cross section showing terms and operation.

Refrigerant is drawn from the evaporator when the secondary pressure drops below the vapour pressure at the evaporator temperature (figure 3). The evaporator flow is drawn into the annular space between the primary jet and the ejector mixing duct wall. The diameter of this duct is carefully chosen such that sufficient condensing pressure may be achieved. The secondary flow sees a converging duct formed by this annulus and is also designed to be choked at optimal operation. Thus the secondary flow can be approximately calculated from the area of the annulus assuming choked flow conditions.

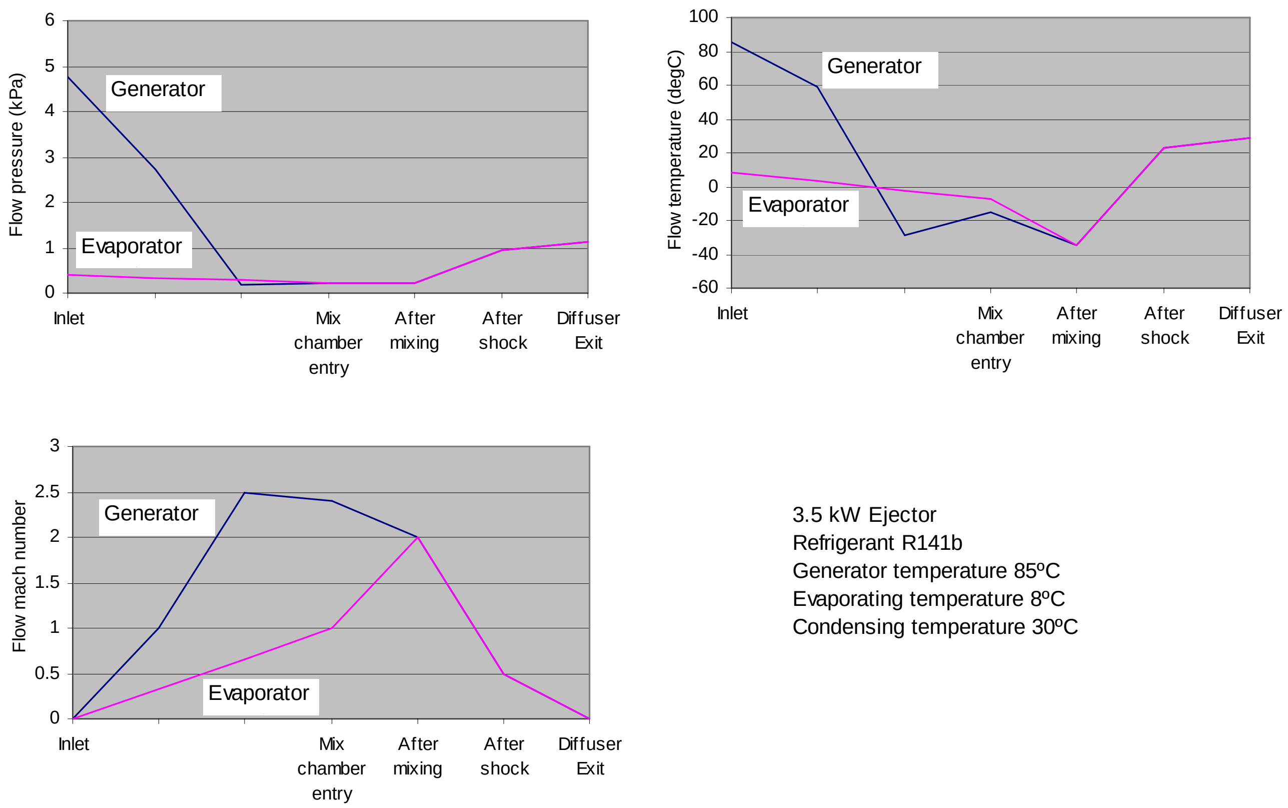

Fig. 3: Pressure, temperature and velocity profiles through a hypothetical ejector

The primary and secondary flows undergo turbulent shear mixing to form a single flow with properties determined by the conservation equations for mass, momentum and energy. The mixed flow is supersonic and unstable with respect to the condensing conditions. It undergoes an irreversible supersonic compression shock which raises its temperature and pressure. This is the second ejector effect, that of compression relative to the evaporator state. Finally, a subsonic diffuser further increases the pressure and temperature by recovering enthalpy from the kinetic energy to allow sufficient temperature to reject heat at condensing conditions.

Development of Ejector Models

Early ejectors were designed by empirical rules. Although a number of empirical models have since been proposed for ejectors (Huang and Chang 1999, Chou et al 2001), a thorough understanding of ejector workings can only be obtained by detailed thermodynamic modelling. Many researchers use the Engineering Sciences Data Unit (ESDU, 1999) published program to design ejectors. This program is partly based on analytical equations and partly upon practical experience, but has fallen behind the latest modeling techniques. Furthermore, this software is also limited to ideal gases, liquids or steam as the ejector fluid.

It wasn't till basic one dimensional ejector theory was proposed by Keenan (1950) that some understanding of the supersonic thermodynamics of ejectors was obtained. These models assumed ideal compressible gas behaviour, ignoring friction and heat loss effects. These models were later refined by others (Huang et al 1999) to compensate for these deficiencies and agreement with experimental results was generally within ±10%.

Model development had focused on the mixing process with Keenan and Newman proposing a constant pressure mixing which was then clarified by Munday and Bagster (1977). In recent developments, Zhu et al (2007) proposed that the secondary flow was two dimensional and axis symmetric, in that the secondary velocity varied radially between the ejector wall boundary layer and the primary flow shear mixing zone. This development, known as the shock circle method, led to marked improvement in model agreement with experimental results. Zhu et al (2008) recently refined this model so that only two parameters require measurement in order to predict ejector performance, although some parameter identification is required. It is fair to say that modeling of the mixing of the generator and evaporator streams in the mixing chamber of the ejector is still evolving.

As an alternative approach to analytical and empirical modeling, some effort has been put into computational fluid dynamics (CFD) models of ejectors. These models have improved recently (Smith et al (1997), Riffat 1999, Rusly et al, 2005) with increases in computing power but are still limited by a lack of fundamental understanding of how the turbulent supersonic mixing takes place within the ejector and how the supersonic boundary layers interact with the ejector walls, particularly around the normal shock in the mixing chamber. The standard κ-ε turbulence models are not good at describing the turbulent primary expansion jet and work continues to find better models.

Based on a better understanding of ejector principles, researchers have proposed a number of improvements to ejector design. The most important are:

1. The replacement of steam as the refrigerant with alternative refrigerants which usually gave superior performance and reduced the size of the ejector
2. The constant rate of momentum change (CRMC) design method proposed by Eames (2004) with improved compression effect and entrainment of evaporator flow
3. Hybrid ejector designs (Sokolov 1993, Huang et al, 2000)) which allow the ejector to continue to operate without sun or to operate at improved COP.

All ejector research published, with one notable exception (Sun, 1996), has concerned ejectors with fixed geometry. In this paper, geometry refers to the ejector mixing duct diameter and ejector length. Ejectors in which these parameters are not constant may be referred to as variable geometry ejectors, although no real examples of such devices are known to exist.

Performance Measures for Ejectors

A key performance parameter of the ejector is the entrainment ratio. This is a ratio of the entrained evaporator flow to the generator flow (equation 1).

$$\omega = \frac{\dot{m}_{ev}}{\dot{m}_g} \qquad (1)$$

Entrainment ratios typically range from 0.15 to over 1.0 for a solar powered ejector system (Huang, 1999). The cooling provided is proportional to the refrigerant mass flow through the evaporator and the enthalpy of evaporation of the refrigerant. A high entrainment ratio is therefore desirable because it means that a higher amount of evaporation/cooling is achieved with a smaller amount of thermal energy.

The entrainment ratio itself is perhaps less useful than the thermal coefficient of performance (COP_{th}) of the ejector, defined as the ratio between the heat removal at the evaporator and the energy input into the cycle through the generator (equation 2). Note that the COP_{th} is related to the entrainment ratio by the thermodynamic properties of the refrigerant.

$$COP_{th} = \frac{Q_e}{Q_g + W_{mec}} \approx \omega \frac{\Delta h_g}{\Delta h_{ev}} \qquad (2)$$

However, for the ejector system where the marginal cost of solar energy is low, this COP does not provide a valid economic comparison with a conventional vapour compression system. It is therefore more relevant to define an electrical COP by equation 3.

$$COP_{el} = \frac{Q_e}{W_{mec}} \quad (3)$$

In this paper, the thermal COP is used to compare fixed and variable geometry ejectors since the comparison is between two thermally driven systems.

FIXED GEOMETRY EJECTORS

The operational characteristics of fixed geometry ejectors are noted by a constant capacity region, a critical operating point and a malfunction region, for a given evaporation and condensing temperature (figure 4). Ideal operation of the ejector, indicated by maximum entrainment of the evaporator flow, is indicated by the knee of each curve in the figure. This point is very close to the malfunction condensing temperature where the entrainment falls to zero so that there is no cooling effect. Indeed the ejector is so sensitive to backpressure (itself related to ambient temperature), that complete malfunction occurs with several degrees of condensing temperature from the optimum operating point.

A second important observation is that an increase in generator (solar) temperature will allow continued operation at elevated condensing temperature but at the expense of COP. This is because the mass flow of the choked primary choked nozzle decreases with increasing driving temperature and thus, there is less motive power to combat the increased condenser backpressure. This implies that a fixed geometry ejector will not be able to take advantage of high collector temperatures during periods of high insolation.

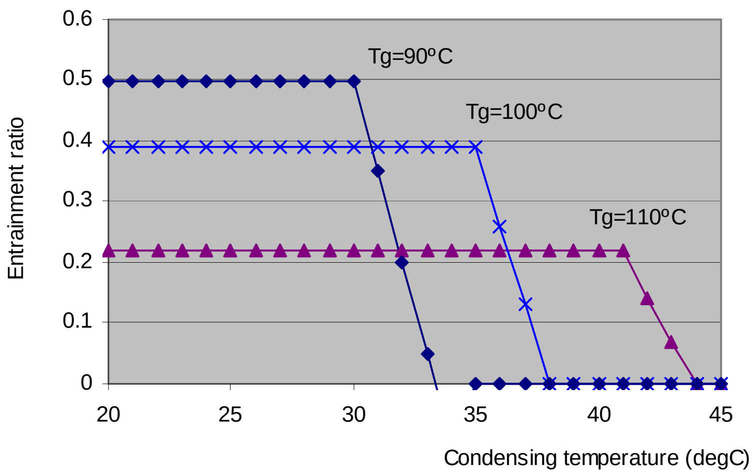

Fig. 4: Fixed geometry ejector operating at a given evaporator temperature (8°C) and generator temperatures, but varying condenser temperature

Loci for a range of optimal operation points for a range of evaporator and generator temperatures can be joined to form a continuous operating curve for a given fixed ejector geometry (figure 5). A number of variations of this map can be produced for practical ejectors to show the effect of various degrees of superheat on the generator, evaporator and condenser flows. Performance maps can readily be configured to show evaporator power, generator power, condenser load, entrainment ratio and COP as a function of generating temperature, evaporating temperature and condensing temperature.

Control of a conventional ejector is thus driven by the ambient temperature which in turn determines the condensing temperature. One may then trade-off the evaporating temperature against available solar collector temperature to determine optimal operation. The practical method of control may then be reduced to a simple lookup table within the memory of a compact controller.

In general, it is best to operate at the lowest generator temperature possible which still allows sufficient condensing pressure, since the collector efficiency drops with increasing collector temperature. Although an increase in evaporator temperature allows substantial gains in cooling power for a given collector temperature, this technique does not allow much lower generator temperatures to be used.

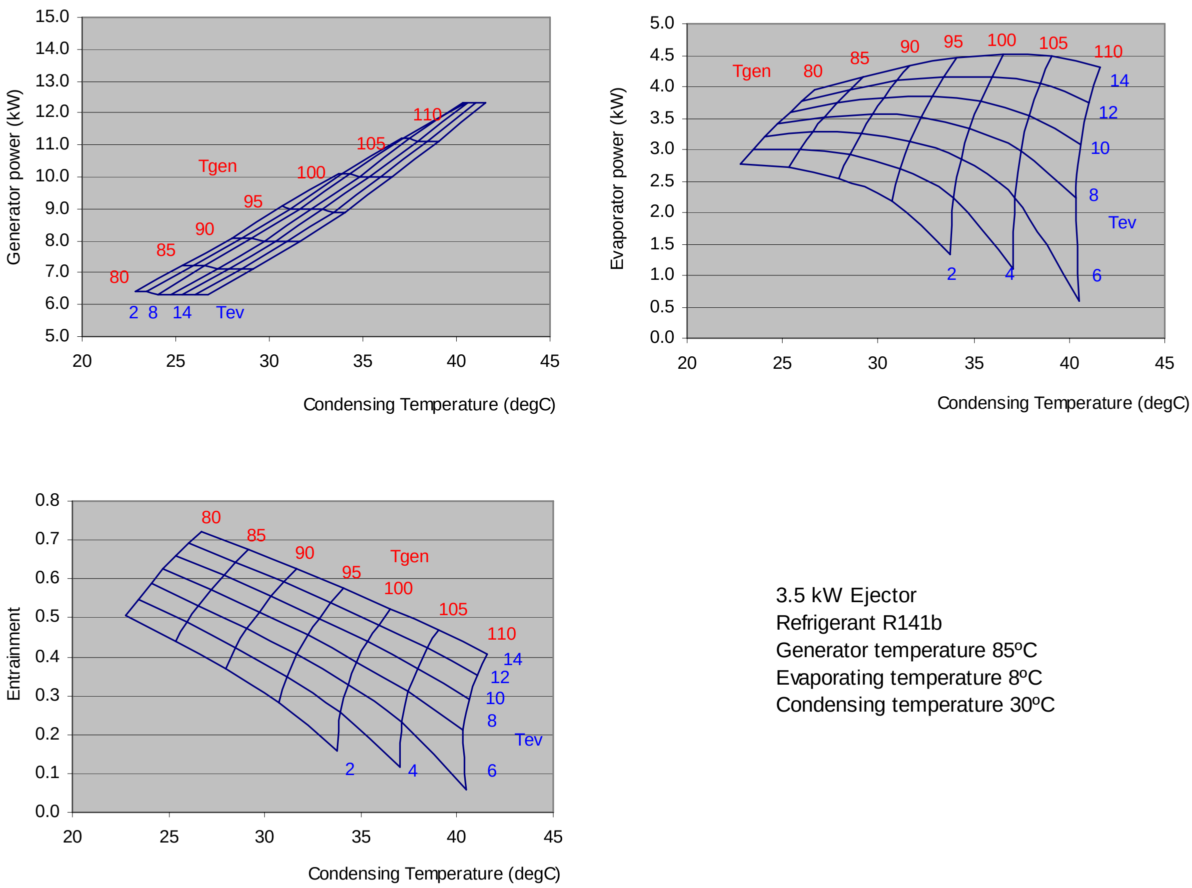

Fig. 5: Performance maps for a fixed geometry ejector

A number of studies (Arbel (2004), Huang (2001)) have investigated the compromise between collector cost and ejector performance although only one has critically examined the coupling between solar collectors and ejector coolers in simulated conditions (Dennis 2009). This study uses the performance map approach previously outlined but fails to account for real-time dynamics of the ejector-collector coupling.

Over the history of fixed geometry ejector development, there have been a number of step changes in performance of ejector cooling systems. The first change came from substituting steam for other refrigerants. Despite its high latent heat, steam has a large specific volume and a very low vapour pressure at low temperatures, leading to a low entrainment ratio and COP.

The second big change in performance, proposed by Sokolov (1993) was to couple an ejector with a conventional heat pump through a common heat exchanger. The arrangement provides improved COP for both the heat pump and the ejector as well as providing cooling at times when there is no sun. The configuration was tuneable such that the size of the solar collector could be traded off against the electricity consumption to optimise the mix of capital and operating costs.

The third and latest step change in performance was proposed by Eames (2004) with the constant rate of momentum change (CRMC) ejector design. The normal shock in the mixing chamber was eliminated along with the loss in stagnation energy and large gain in entropy. The ejector pressure lift was shown experimentally to improve by up to 50% and the ejector was capable of operating at higher compression ratios, thus at higher ambient temperatures.

Over the last twenty years, there have been incremental changes in ejector performance through improved heat recovery, clever designs of primary nozzles and various hybrid systems.

Perhaps the next step change is to be made using variable geometry ejectors. Sun (1996) produced modelling which speculated that a variable geometry ejector might possibly provide the next step change in ejector performance, but did not fully evaluate the potential or produce a design for a practical device. Sun suggested that such an ejector would require a varying area ratio and length in order to work optimally over a range of generator and condenser temperatures. The paper proposed an iterative equation to find the ejector area ratio based on condenser and generator temperature.

VARIABLE GEOMETRY EJECTORS

Once the operating temperatures for the generator, evaporator and condenser are known, the operating point for a fixed geometry ejector is found by manipulating the evaporator mass flow rate such that the required condensing pressure is met, assuming the required level of superheating of each flow. Thus maps for the entrainment ratio, COP and evaporating power (cooling effect) may be readily obtained by repeating this procedure across the range of temperatures of interest.

For variable geometry ejectors, the secondary duct diameter is variable and thus the area ratio can be changed. This removes the constant capacity constraint of the fixed ejector, and thus additional generator temperature leads to increased entrainment at temperatures below the knee condensing temperature provide that a larger diameter mixing duct is available. Additional generator temperature also allows the ejector to operate at higher condensing temperatures by restricting the diameter of the mixing chamber.

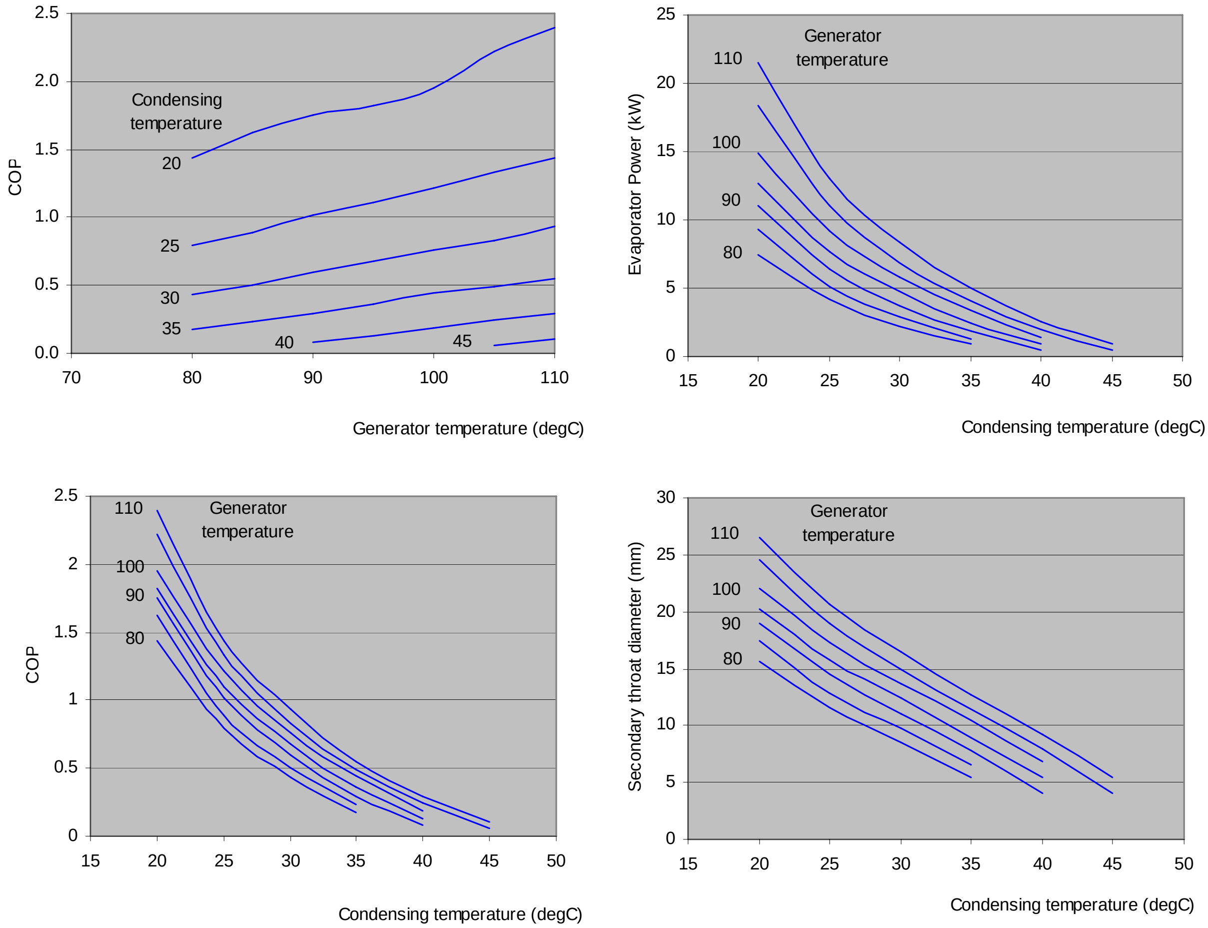

Fig. 6: Performance maps for a variable geometry ejector with fixed evaporating temperature 8ºC

The variable geometry ejector performance plots (figure 6) show that the variable geometry ejector operates rather differently to the fixed geometry ejector. It's COP rises with generator temperature for all condensing temperatures. It is thus able to utilize the full capacity of a solar collector when high temperatures are available. Note that the COP is equivalent to the fixed ejector at the design condition of generator , evaporator and condenser temperatures of 95ºC, 8ºC and 32ºC respectively.

It is immediately obvious that the variable geometry ejector has a great deal of additional capacity at low condensing temperatures. Since this condition relates to ambient conditions where cooling may not be required, these characteristics hint that variable geometry ejectors coupled to storage might be a productive area of further research (Dennis 2009). Alternatively, the additional capacity might be used to cool the thermal mass of the house during the morning in preparation for a warm afternoon, although a house should not be regarded as an efficient store of coolth.

Another notable feature, first noted by Sun, was the less severe decline in ejector performance at elevated condensing temperature. Comparison of figures 5 and 6 indicate that there is no knee point on a variable geometry ejector performance curves and thus control of the primary flow is less critical.

Thus the variable geometry ejector favours higher collector temperatures and the control map would operate the ejector at the highest generator temperature that can be supported by the collector at the prevalent conditions. The ejector mixing duct area ratio is manipulated such that the required condensing pressure is met, again assuming the required level of superheating of each flow. Since the geometry of the primary jet is known, the annular space between the primary jet and the wall forms the choking nozzle for the secondary jet and hence the evaporating mass flow and related quantities can be determined from the control maps.

VARIABLE GEOMETRY EJECTOR DESIGN

The variation in the mixing duct diameter, also shown in figure 7, indicates that a range of mixing duct diameters spanning 8mm to 22mm would be sufficient to operate a variable geometry ejector satisfactorily over the range of condensing temperatures indicated. To date, there have been no published or patented designs for variable geometry ejectors.

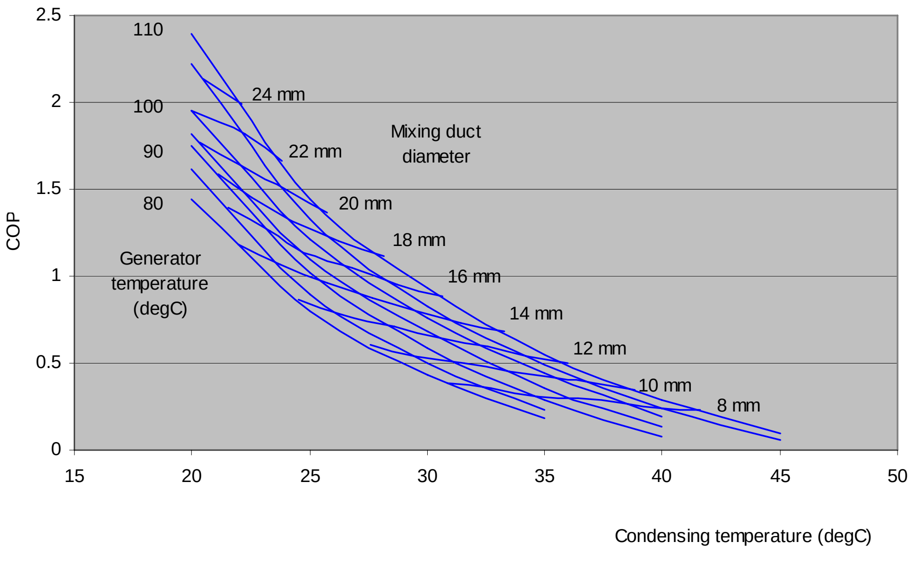

Fig. 7: Performance map for a variable geometry ejector, showing the required mixing duct diameter

CONCLUSION

There is clearly a need to reduce peak electricity consumption in Australia and this may be achieved in part by offsetting electrical compression in popular vapour compression residential air conditioners.
Fixed geometry solar powered ejector air conditioning systems are very sensitive to condensing and evaporating conditions in particular. This limits their operational flexibility and hence their commercial viability. There are a number of ways in which solar ejector systems can be improved. The most promising approach seems to be a retrofit option involving a conventional vapour compression system coupled to a solar powered ejector system utilising smart control and possibly thermal storage. Modelling and experimental studies are underway at the Australian National University to evaluate a number of possible improvements to the flexibility of application of these devices.

LIST OF SYMBOLS

Symbol	Units	Description
T_{gen}	°C	*Generator vapour outlet temperature*
T_{ev}	°C	*Evaporator vapour outlet temperature*
Q_{gen}	*W*	*Generator power consumption*
Q_{ev}	*W*	*Evaporator cooling load*
m_{gen}	*g/s*	*Generator refrigerant mass flow*
m_{ev}	*g/s*	*Evaporator refrigerant mass flow*
W_{mec}	*W*	*Electrical work input to system (pumps)*

REFERENCES

1. Huang, B., Chang, J., (1999), *Empirical correlation for ejector design,* International Journal of Refrigeration, 1999, 22:379-388.

2. Chou, S., Yang, P., Yap, C. (2001), M*aximum mass flow ratio due to secondary flow choking in an ejector refrigeration system,* International Journal of Refrigeration, 2001, 24:486-499.

3. Engineering Sciences Data Unit (ESDU), *Ejectors and Jet Pumps: Design and Performance for Compressible Gas Flow,* ESDU data item 92042, Issue 2007-01, ESDU International Ltd.

4. Keenan J., Newman E., *A simple air ejector*, Journal of Applied Mechanics Transactions, ASME 1942:64:A75-81

5. Huang B., Chang J., Wang C., Petrenko V., *A 1D analysis of ejector performance*, International Journal of Refrigeration, 1999, 22:354-364

6. Keenan J., Newman E., Lustwerk F. *An investigation of ejector design by analysis and experiment,* Journal of Applied Mechanics Transactions, ASME 1950:72:299-309

7. Munday J., Bagster D., *A new ejector theory applied to steam jet refrigeration*, Ind Eng Chem Process Des Dev 1977:16:442-449

8. Zhu Y., Cai W., Wen C., Li Y., *Shock circle model for ejector performance evaluation*, Energy Conversion Management, 2007:48:2533-2541

9. Zhu Y., Cai W., Wen C., Li Y., *Simplified ejector model for control and optimisation*, Energy Conversion and Management, Vol 49, 2008, 1424-1432.

10. Smith S., Riffat S., Wu S., Eames I., *Low pressure ejectors: prediction of performance by CFD*, Building Services Engineering, Residential Technolgy, 1997 18(3):179-182

11. Riffat S., Everitt P., *Experimental and CFD modelling of an ejector system for vehicle air conditioning,* Jourrnal Inst Energy, 1999, 72:41-47

12. Rusly E, Aye L., Charters W., Ooi A., *CFD analysis of ejector in a combined ejector cooling system*, International Journal of Refrigeration 2005, 28:1092-1101

13. Eames I., *A new prescription for the design of supersonic jet pumps: the constant rate of momentum change method*, Applied Thermal Engineering, Vol 22, 2002, 121-131

14. Sokolov, M., Hergshal, D. (1993), *Optimal coupling and feasibility of a solar powered year round ejector air conditioner,* Solar Energy, 50 (6), 507-516.

15. Huang B., Petrenko V., Chang J., Lin C., Hu S., *A combined cycle refrigeration system using ejector cooling cycle as the botom cycle*, International Journal of Refrigeration, 2000, 24:391-399

16. Sun, D. (1996), *Variable geometry ejectors and their applications in ejector refrigeration systems*, Energy, 21 (10), 919-929

17. Arbel A., Sokolov M., *Revisitng solar powered ejector air conditioner – the greener thebetter*, Solar Energy 2004, 77:57-66

18. Huang B., Petrenko V., Samofatov Y., Shchetinina N., *Collector selection for solar ejector cooling system,* Solar Energy 2001, 71:269-274

19. Dennis M., Garzoli K., *Modelling of variable geometry solar assisted ejectors with cold store,* Proceedings of Eurotherm 85, Belgium, 2009

BRIEF BIOGRAPHY OF PRESENTER

Dr Mike Dennis is a senior research fellow with the Centre for Sustainable Energy at the Australian National University. He has ten years of international experience with large scale industrial process control and information systems. After completing a PhD in smart control of solar hot water systems in 2004, he has continued to develop renewable energy technologies for residential applications.

His chief interest is in solar air conditioning and hybrid residential solar thermal solutions. Other technologies that Mike has contributed to include concentrating photovoltaic/thermal collectors, sun tracking systems, the ANU Big Dish solar thermal collector, micro-encapsulated phase change materials and clathrate cold stores.