COMBINED SOLAR AND ELECTRIC SYSTEM FOR SPACE COOLING

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ABSTRACT

Combined solar ejector-vapour compression air conditioning system comprises an ejector cycle operating at the higher pressure-loop powered by solar energy and a low-pressure conventional vapour compression cycle. The combined system harvests the advantages of both the ejector and vapour compression cycles, therefore provides higher COP than conventional cycles. This system has attractive economic and environmental propositions since it uses solar energy as its major energy source. Most working fluids studied for the combined solar ejector-vapour compression cycle are the HCFCs and HFCs. Due to their high global warming potential (GWP), these types of refrigerants are next in line to be phased out following the CFCs. The focus of this paper is to investigate the potentials of some alternative (and natural) refrigerants such as ammonia, hydrocarbons, carbon dioxide and water. Thermodynamic, economic, environmental and performance comparisons were made with several common existing refrigerants. Combined system performance simulations were performed with the refrigerants used in pairs under varying intercooler temperature. Effects of intercooler temperature variation on system performance were pointed out. Refrigerants responses to changing intercooler temperature were also discussed.

1. INTRODUCTION

The dominance of the mechanical vapour compression systems in HVAC&R industries is unquestionable owing to their high efficiency and low capital cost. However, these systems are powered by mains electricity generated by burning fossil fuels. One of the consequences of this is the release of unwanted greenhouse gases (GHGs) as by-products into the atmosphere. The increased amount of these gases in the earth's atmosphere causes a rise in the absorption of the solar radiation from the sun and in turn warming the earth surface and the lower atmosphere; a phenomenon known as 'Global Warming'. Moreover, the CFC refrigerants used in these systems are harmful to the stratospheric ozone. The HCFC and HFC refrigerants for replacement of CFCs are still expensive and provide only temporary solutions as they are facing a phase out in the future themselves.

Research efforts into alternative systems powered by renewable energy sources with the potential of using new environment-friendly or natural refrigerants have been stronger than ever. Absorption systems are the most popular, not only because they are heat-powered but also they use natural refrigerants such as ammonia-water or lithium bromide-water. Nevertheless, the expensive construction and installation of more components, high maintenance cost and relatively low COP values have been their major drawbacks.

Solar energy can be harvested as thermal energy and its application in air conditioning is very practical since the demand for cooling and the availability of solar radiation are very much in phase. Solar energy usage has received tremendous developments in the past decade, making its cost and energy ratio more and more affordable. A system powered by solar energy yet uses ‘green’ refrigerants seems to be a perfect combination. Absorption cycles have been studied using thermal energy from this source but have not performed cost-effectively.

An ejector refrigeration cycle is another alternative system that runs primarily on thermal power. Apart from a small amount of mechanical work needed for the circulating pump, the cycle can be thermally powered by inexpensive and environment-friendly energy sources such as solar energy, waste heat, natural gas, geothermal energy, etc. Despite its simplicity and reliability, ejector cycles powered by solar energy are still not economically interesting since experiment data shows very low COPs, with typical values of 0.2. However, recent studies show that when used in combination with other cycles, such as absorption or vapour compression cycles, the overall system performance yields improvement over the individual conventional cycles (Aphornratana & Eames 1998, Sun 1997).

An ejector refrigeration cycle is similar to a mechanical vapour-compression cycle except that an ejector, a liquid pump and a generator are used in place of a compressor. The working principle of an ejector cooling process is as follows. Pressurised primary working fluid boiled off at the generator, then enters and expands through the ejector primary nozzle. As induced by the pressure difference between the condenser and the generator, this stream is accelerated, choked at the nozzle throat and expanded into a low-pressure supersonic flow (pressure energy to kinetic energy). This flow at the nozzle exit will entrain the secondary flow from the evaporator. The two streams mix at supersonic speed in the mixing chamber and further down the duct, a normal shock wave will reduce the mixed flow’s velocity to a subsonic value. It then enters the subsonic diffuser where it undergoes a re-compression process to reach the condenser pressure at near zero velocity (kinetic energy to pressure energy). When flowing into the condenser, the mixed flow condenses into liquid. This liquid is finally divided into two parts, one goes to the evaporator and the other is pumped back to the generator.

Ejector systems offer the most simple and reliable technology. They feature low installation and low operating costs. There is no moving part except a small circulating pump thus produces low noise and vibration. Mechanical vapour compression has high efficiency and low capital cost. It has established its solid identity through decades of development. Bringing together the advantages of these two systems in a combined system and introducing solar energy to power the ejector cycle have been an attractive option and theoretical analysis shows this combined system can give a 20-50% improvement over the individual sub-cycles that comprise it (Sun 1997, Sokolov & Hershgal 1993b).

The combined system consists of an ejector cooling sub-cycle operating at around 25-100°C range and a vapour compression sub-cycle operating at around 5-45°C range. The two cycles meet at the intercooler which functions as an evaporator for the ejector cycle and a condenser for the compression cycle. In the intercooler, refrigerant vapour from the vapour-compression side needs to reject heat to liquid refrigerant from the ejector side therefore there must be a temperature difference between the two. This temperature difference is normally 5-8°C and a function of the heat transfer area of the heat exchanger (intercooler). For increased efficiency, regenerators for heat recovery can be added between the condenser and the ejector to pre-heat the liquid refrigerant pumped to the generator. The schematic diagram of the combined system can be depicted in the figure below.

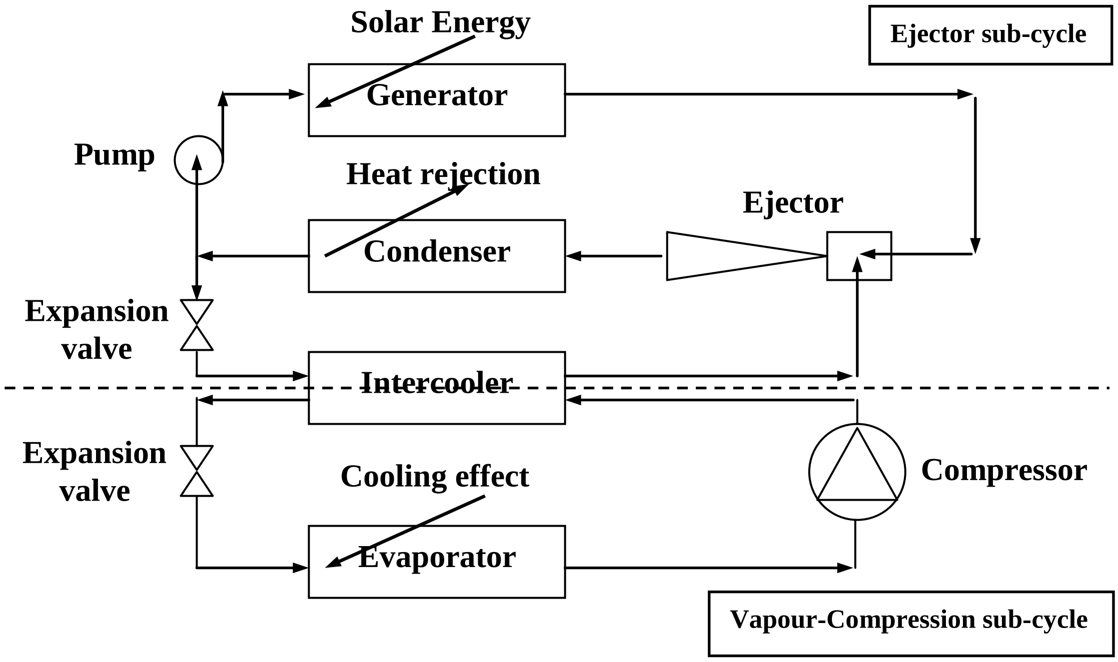

Figure 1. The solar powered combined ejector-vapour compression cycle.

Recent studies for this combined system concentrate on gaining overall theoretical improvement using refrigerants from HCFC and HFC such as R134a, R114, R123, R141b, etc. and understanding the behaviour of the system under varying operating conditions. Investigation on the characteristics of refrigerants from the physical & thermodynamic, chemical, environmental, economic and system requirements point of views is still lacking. Relevantly, alternative refrigerants are being intensively introduced into refrigeration systems especially because HCFCs and HFCs are only temporary substitutes. Therefore, it is the focus of this study that we analyse the sub-cycles and combined system refrigerant criteria, assess, test and select several new (or preferably natural) and alternative refrigerants that not only have potential for long term and environment-friendly use in the combined system but also offer reasonable overall COP. The refrigerants selected could be used as a single refrigerant or as in this paper, a pair of different refrigerants each tailored for the different sub-cycles. Also pointed out are the effects of intercooler temperature variation on system performance for a given operating condition. A discussion on different responses of refrigerants towards certain changes in the system parameters is also given.

2. WORKING FLUID CRITERIA

The criteria for an ideal refrigerant for use in the combined solar powered combined ejector-vapour compression are as follows:

1. Type of refrigerant

It is preferable that the refrigerant(s) is environmentally friendly. The best option is a natural refrigerant such as ammonia, water, CO_2, etc.

2. Physical properties; critical temperature and critical pressure

The temperature operating range of the solar collector to be economically feasible is around 80-100°C. This can be easily achieved by an advanced honeycomb flat-plate solar collector with reasonable cost. The critical temperature of the refrigerant has to be higher than the generator temperature to allow phase change during vaporisation at the generator. In contrast, the critical pressure of the refrigerant has to be low to minimise pump work and ensure good performance at low condensing pressure.

3. Environmental impact and Safety

The refrigerant must have zero ODP (Ozone Depleting Potential) and low or close to zero GWP (Global Warming Potential). In addition, it must have none or the lowest possible flammability and toxicity for safety reasons.

4. Thermodynamic characteristics

The latent heat of vaporisation is an important property. At the generator on the ejector sub-cycle, this property has to be optimally high to effectively absorb as much solar energy generated by the solar collector as possible with a small mass flow rate. However, at the evaporator on the vapour compression side, this property is expected to be reasonably low to maximise the efficiency of the cycle.

The specific volume has to be low to minimise the size and cost of the piping and compressor. Thermal conductivity is expected to be high to facilitate heat exchange at the heat exchangers. Viscosity is expected to be low to have a low friction flow coefficient during the refrigerant being distributed in the pipes.

5. Slope of saturation vapour line

For the refrigerant to experience no condensation during isentropic expansion of the vapour refrigerant in the nozzle of the ejector, it has to have a positive slope saturation line in its T-s diagram at ejector cycle operating range. For the vapour-compression sub-cycle, a refrigerant with negative slope is preferable to guarantee problem free compression at the compressor without the need of superheating.

6. Performance and Efficiency

For the ejector sub-cycle, performance of a refrigerant will be measured from its ejector entrainment ratio (ratio of the secondary and the primary mass flow rate) and area ratio (ratio of the constant cross section area and the throat area of the ejector's primary nozzle). Entrainment ratio determines the refrigerating effect gained at the evaporator through the amount of secondary flow entrained by the ejector while area ratio determines the physical dimensions of the ejector. And for the mechanical vapour-compression sub-cycle, performance of a refrigerant is determined from the COP of the cycle i.e. the ratio of the useful refrigerating effect to the power/work done to produce it.

For the combined system, performance can be measured in terms of COP mechanical and COP thermal as defined as follows:

$$COP_{me} = \frac{Q_e}{W_c + W_p} \; ; \; COP_{th} = \frac{Q_e}{Q_g + W_c + W_p}$$

The COP mechanical is the ratio of useful cooling (cooling effect at the evaporator, Qe) and the electrical power consumption (work by compressor, Wc and pump, Wp) and the COP thermal is the ratio of Qe and the thermal heat supplied at the generator plus the work done by the compressor and the pump.

7. Cost and Availability

Definitely the intention is to use a refrigerant with low cost and good availability.

8. Compatibility

The refrigerant used has to be compatible with the materials and lubricants in contact with it.

3. METHODOLOGIES

The main objective of this paper is to explore the potentials of several alternative refrigerants with reference to the criteria above for use in the combined ejector-vapour compression cycle. As a starting point, a table listing refrigerant properties in relation to the criteria given above was constructed for the proposed refrigerants and several other existing ones (Table 1.). Following on is a simulation to select the best refrigerant or refrigerant pair from the proposed group that best satisfies the requirements above. In the simulation, each refrigerant was simulated as being used in the ejector sub-cycle and all the other refrigerants were set into the vapour compression sub-cycle one at a time and the thermodynamic analysis of the combined cycle was performed to calculate the overall COP of the system.

The physical and thermodynamic properties of the refrigerants were obtained from REFPROP (NIST standard reference database 23 version 6.01) (McLinden et al. 1998). The environmental impact, cost and compatibility associated with the refrigerants were based on the data in ASHRAE Fundamentals 1997 and Refrigeration and Air Conditioning of Air Conditioning and Refrigeration Institute. Performance results for ejector sub-cycle follow the calculation methods developed by Huang et al. (1999). The overall COPth and COPme in the combined ejector-vapour compression system were derived following the thermodynamic analysis in Sun (1997).

In the ejector model, several coefficients accounting for losses in the nozzle and constant area section were taken as in Huang et al. (1999). These coefficients are:

η_p; coefficient relating to isentropic efficiency of the compressible flow in the nozzle

η_s; coefficient relating to isentropic efficiency of the entrained flow

ϕ_p; nozzle exit loss coefficient for the primary flow

ϕ_m; coefficient accounting for the frictional loss in the cross section area

The values used for η_p, η_s, ϕ_p are constants of 0.95, 0.85 and 0.88, respectively. These coefficients are related to the ejector design and manufacturing technique and the figures used above have been shown to give appropriate fit with the experimental data. ϕ_m was taken as 0.9 for all the refrigerants tested to set a consistent basis for comparisons.

In the combined cycle thermodynamic analysis, a temperature difference of 6°C was employed for the two fluids that exchange heat at the intercooler. This means the condensing temperature of the vapour-compression sub-cycle is 6°C higher than the evaporating temperature of the ejector sub-cycle.

Also studied is the effect of intercooler temperature on the overall system performance in terms of COPth and COPme. This is performed by varying intercooler temperature under two sets of different generator and condenser temperatures. The results will pave the way for determining the intercooler temperature for optimum overall system performance.

Refrigerants considered are from the natural types, hydrocarbon, HFC and other alternatives in order of priority. From the natural type, we chose to put in the test: ammonia (R717), CO_2 (R744), water/steam (R718), from hydrocarbon: propane (R290), from HFC: R152a, from fluoro propane compound: R245ca, R245fa, from azeotropic compound: R500 and we also included R22, R123, R141b from HCFC and R12 from CFC for comparison purposes.

4. RESULTS AND DISCUSSION

4.1. Single Refrigerant Analysis

Performance indicators in Table 1 presents values for individual cycles as well as combined cycle where the refrigerants are used as a single working fluid. In vapour-compression

Table 1. Properties and performance values for some existing and proposed refrigerants

	Existing Refrigerants					Proposed refrigerants							
Refrigerant	**R12**	**R22**	**R123**	**R141b**	**R142b**	**R152a**	**R245ca**	**R245fa**	**R717**	**R290** **propane**	**R718** **water/steam**	**CO2**	**R500**
Type	CFC	HCHC	HCFC	HCFC	HCFC	HFC	fluoro propane compound.	fluoro propane compound	natural	Hydro-carbon	natural	natural	azeotropic mixture (CFC)
Critical temperature (°C)	111.97	96	183.68	204.20	137.1	113.26	174.42	154.05	132.25	96.7	373.99	31.06	102.1
Critical pressure (kPa)	4136	4974	3674	4250	4123	4492	3925	3640	11417	4248	22064	7372	4423
ODP	1.0	0.05	0.02	0.06-0.08	0.06	0	0	0	0	0	0	0	0.74
GWP 100 year	7300	1500	90	1500-1800	2000	140	560	820	≈0	≈0	0	0	6010
Safety group*	A1	A1	B1	A2	A2	A2	flammable	non-flmbl	B2	A3	A1	A1	A1
Lat. heat of vaporisation (@ 100°C, kJ/kg)	64	--	134	185.1	131.2	132.7	151.5	135.5	716	n/a	2256.6	n/a	73.8 @ 92°C
Specific volume (V @ 20°C, m^3/kg)	0.03112	0.026	0.20354	0.31167	0.07687	0.06267	0.2107	0.1397	0.14923	0.05522	57.78	0.00514	0.03130
Thermal conductivity (L/V @ 20°C, mW/mK)	68.87 / 9.98	85.89 10.95	78.1 / 9.81	93.1 / -	83.19 / 11.17	103.9 / -	88.87 / 11.87	82.56 / 12.08	479.2 / 25.38	95.4 / 18.38	598.4 / 18.23	87.2 / 34.19	77.0 / -
Viscosity (L/V @ 20°C, μPa s	201.4 / 11.57	175.3 12.43	445.8 / -	444.7 / -	254.8 / 9.693	172.6 / 9.94	583.6 / 9.983	431.5 / 10.16	141.6 / 9.71	102.5 / 8.53	1002.1 / 9.73	67.2 / 18.12	199.1 / -
Saturation line slope for ejector cycle	negative	negative	positive	positive	slightly negative	negative	positive	positive	negative	negative	negative	negative	negative
Saturation line slope for vapour compression cycle	negative	negative	negative	negative	negative	negative	positive	positive	negative	negative	negative	positive	negative
COP vapour compression	17.23	--	17.67	17.79	17.5	17.38	17.59	17.49	17.39	16.95	15.64	13.16	17.35
Ejector Entrainment Ratio	0.231	0	0.182	0.182	0.251	0.215	0.168	0.185	0.189	0.089	0.032	--	0.144
Ejector Area Ratio	2.42	1.02	4.08	4.1	3.23	2.56	4.2	4	2.46	1.46	6.25	--	1.85
Combined Ejector-Vapour Compression COP as single refrigerant	0.19	0	0.148	0.151	0.205	0.184	0.134	0.146	0.174	0.074	0.029	--	0.122
Cost & availability	phased out	mdt/ good	high/ poor	high/poor	high/poor	very high/poor	high/poor	high/poor	low/ good	low/good	very low/ good	low/good	high/poor
Compatibility	moderate	good	moderate	good	good	good	good	good	poor	good	good	moderate	good

sub-cycle, R245ca, R245fa, R717 and R500 are comparatively as good as the most efficient refrigerants from the existing group, i.e. R123 and R141b.
As for the ejector sub-cycle, R152a has the highest performance indicator (entrainment ratio), followed by ammonia (R717), R245fa and R245ca. As a single refrigerant in the combined cycle, R152a is still the most efficient with the highest COPth. Ammonia is second, followed by R245fa and R245ca. The R142b from the existing group has the highest COPth of all. CO_2 (R744) has very low vapour compression COP and is unsuitable for the ejector sub-cycle due to its low critical temperature. Note that all these figures are snapshot calculations at fixed generator, condenser, intercooler and evaporator temperatures of 90, 50, 20 and 5°C, respectively.

4.2. Dual Refrigerant Analysis

Fig. 2 shows the COPth and COPme of the overall best combinations of refrigerants for the given intercooler temperatures at generator temperatures of 90 and 80°C (with condenser and evaporator temperatures fixed at 50 and 5°C, respectively). It is evident that COPth increases as the intercooler temperature increases, but the opposite goes for COPme. Generator temperature has an important effect on COPth in that it increases COPth when raised. However, COPme seems to be impervious to it. Since COPth and COPme have opposite trends to increasing intercooler temperature, it is possible to find an optimum intercooler temperature for the given operating conditions.

Fig. 3 has exactly the same explanation as Fig. 2, except that the condenser temperature is fixed at 40°C. The result of this is higher values for both COPth and COPme. Therefore, the condenser temperature can only give positive effects on the system performance as it decreases. Unfortunately, to ensure heat exchange, the condenser temperature of air-cooled condensers is limited by the ambient temperature.

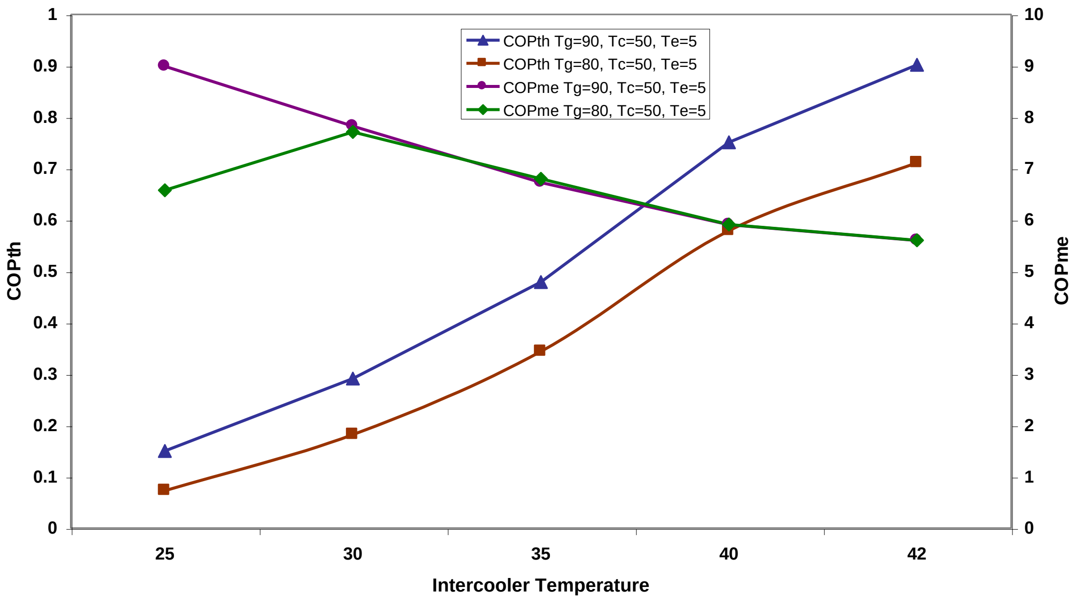

Figure 2. COPth and COPme of the overall best refrigerant combinations under varying intercooler temperature (Tc = 50 and Te = 5).

Table 2. COPth and COPme of the overall best refrigerant combinations and the proposed refrigerants' best combinations at 50°C condenser temperature, 5°C evaporating temperature.

Int. temp	Overall best pair	COPth	COPme	Best from proposed group	COPth	COPme
Tg = 90, Tc = 50, Te = 5						
25	R141b/R123,R141b, R142b,R245ca	0.15	8.99	R245fa/R141b	0.142	8.27
30	R141b/R123,R141b	0.291	7.83	R245fa/R141b	0.276	7.56
35	R141b/R123,R141b	0.479	6.73	R245ca/R141b	0.462	6.74
40	R141b/R141b	0.751	5.91	R245ca/R141b	0.733	5.88
42	R141b/R141b	0.902	5.60	R245ca/R141b	0.884	5.58
Tg = 80, Tc = 50, Te = 5						
25	R142b/R123,R141b, R142b,R245ca	0.073	6.58	R245fa/R141b	0.059	7.45
30	R141b/R123,R141b, R142b,R245ca	0.182	7.71	R245fa/R141b,R123	0.175	7.45
35	R141b/R141b	0.344	6.80	R245ca/R141b	0.334	6.75
40	R141b/R141b	0.579	5.91	R245ca/R141b	0.571	5.89
42	R141b/R141b	0.711	5.60	R245ca,R245fa/R141b ;R718/R141b	0.7 ;0.685	5.58 ;5.08

Table 2 summarises the performance values in terms of COPth and COPme for the most efficient refrigerant combinations of all as opposed to those from the proposed group for the given varying intercooler temperatures at generator temperature of 90 and 80°C and fixed condenser and evaporator temperatures of 50 and 5°C, respectively. Note that the refrigerant(s) at the left side of the backslash is the refrigerant used in ejector sub-cycle whereas the refrigerant(s) on the right side is the refrigerant used in the vapour compression sub-cycle.

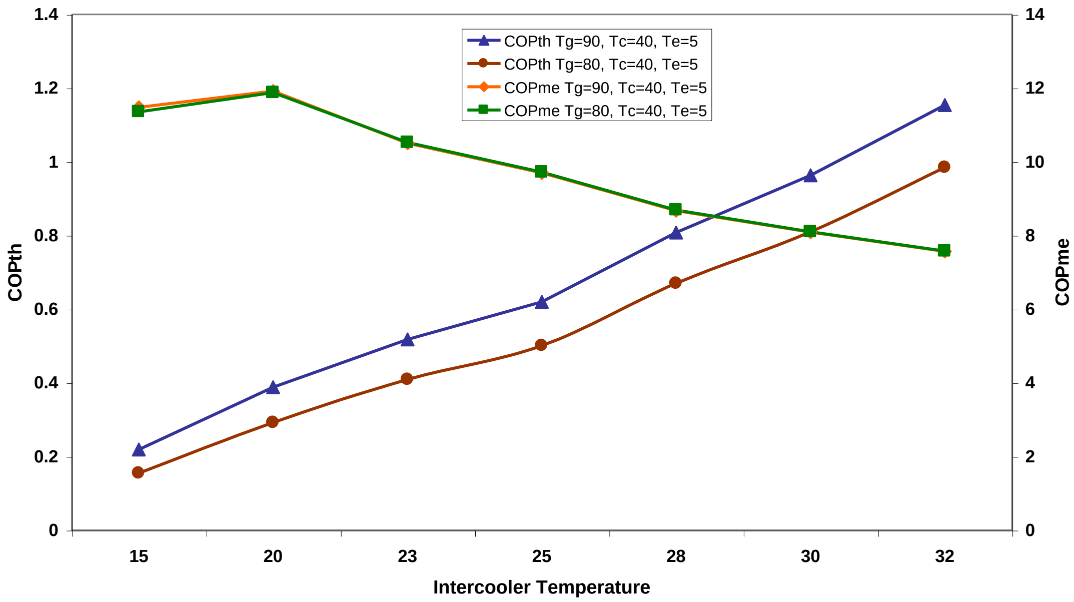

Figure 3. COPth and COPme of the overall best refrigerant combinations under varying intercooler temperature (Tc = 40 and Te = 5).

Table 3. COPth and COPme of the overall best refrigerant combinations and the proposed refrigerants' best combinations at 40°C condenser temperature, 5°C evaporating temperature.

Int. temp	Overall best pair	COPth	COPme	Best from proposed group	COPth	COPme
Tg = 90, Tc = 40, Te = 5						
15	R142b/any	0.219	11.48	R245fa/any	0.203	13.04
20	R141b/R141b	0.388	11.92	R245fa/R123,R141b,R245ca	0.366	11.24
23	R141b/R141b	0.518	10.50	R245ca/R123,R141b	0.493	10.31
25	R141b/R141b	0.62	9.70	R245ca/R141b	0.595	9.61
28	R141b/R141b	0.808	8.68	R245ca/R141b	0.78	8.62
30	R141b/R141b	0.963	8.09	R245ca/R141b	0.936	8.05
32	R141b/R141b	1.154	7.57	R245ca/R141b	1.127	7.54
Tg = 80, Tc = 40, Te = 5						
15	R142b/any	0.155	11.36	R245fa/any	0.131	12.74
20	R141b/R123,R141b	0.292	11.88	R245fa/R123,R141b,R245ca	0.28	11.33
23	R141b/R141b	0.409	10.53	R245ca,R245fa/R123,R141b	0.392	10.26
25	R141b/R141b	0.501	9.72	R245ca/R141b	0.484	9.64
28	R141b/R141b	0.67	8.69	R245ca/R141b	0.653	8.64
30	R141b/R141b	0.81	8.10	R245ca/R141b	0.794	8.07
32	R141b/R141b	0.985	7.58	R245ca/R141b	0.97	7.55

It can be seen that when R245ca and R245fa are used in the ejector sub-cycle with R141b in the vapour compression sub-cycle, the COPth and COPme resulted are comparatively as good as those of the best overall refrigerant combinations, i.e. R141b/R141b in many cases. Interestingly, at high intercooler temperature, i.e. 42°C for generator temperature of 80°C, water/steam (R718) in the ejector sub-cycle pairing with R141b in the compression sub-cycle has COPth of almost as high as that of R245ca or R245fa paired with R141b.

Table 3 further confirms the good characteristics of R245ca and R245fa when used in ejector sub-cycle combined with R141b (or R123 in some cases) in vapour-compression sub-cycle. The fact that R141b has always outperformed other refrigerants in vapour compression sub-cycle is in good agreement with its highest vapour compression COP given in Table 1. As for the R141b performance values in ejector sub-cycle and combined system in Table 1, R141b does not perform particularly well at 20-25 intercooler temperature range, but does so progressively better at 25-42 at a given 50°C condenser temperature. Note that R245ca and R245fa display similar characteristics as well. Refrigerant responses to changing system parameters are discussed in the following section.

Figure 4 and 5 below illustrate the responses of each of the refrigerants tested under varying operating conditions. Figure 4 is the plot of COPth of all the refrigerants at fixed generator temperature of 80°C, condenser temperature of 40°C and evaporator temperature of 5°C. The intercooler temperature is varied incrementally from 15 to 32°C. As evident, different refrigerant responds differently to the increase of intercooler temperature. The slowest responses are recorded by R22, propane and R152a whereas the fastest responses are from water/steam (R718), R141b, R245ca, R123 and R245fa. This behaviour is largely related to the increase quantity of the refrigerant absolute pressure as the intercooler temperature increases. As pressure is directly proportional to the mass flow through the nozzle at choking condition (as defined in the gas dynamic equation), the larger the pressure increase quantity, the larger the entrained flow rate. Consequently, as the primary mass flow rate is constant at fixed generator pressure, a higher entrainment ratio is resulted. The end effect is a more progressive increase in the COPth of the overall system.

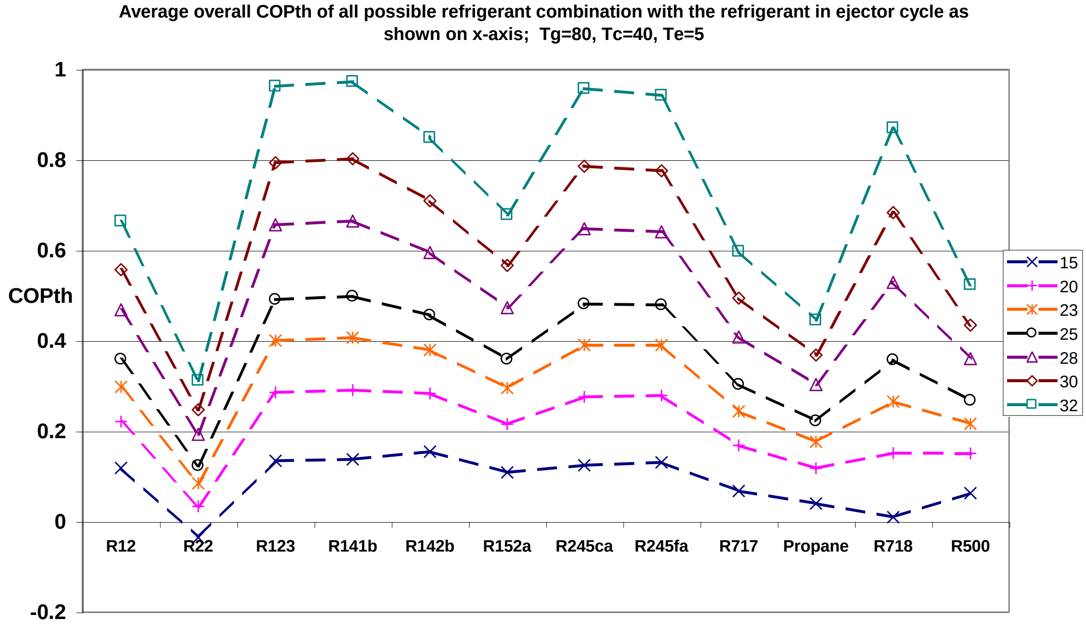

Figure 4. Responses of refrigerants towards changing intercooler temperature. COPth is averaged over combinations of refrigerant on ejector side (x-axis) with the rest, Tg = 80, Tc = 40, Te = 5). Legends show intercooler temperatures.

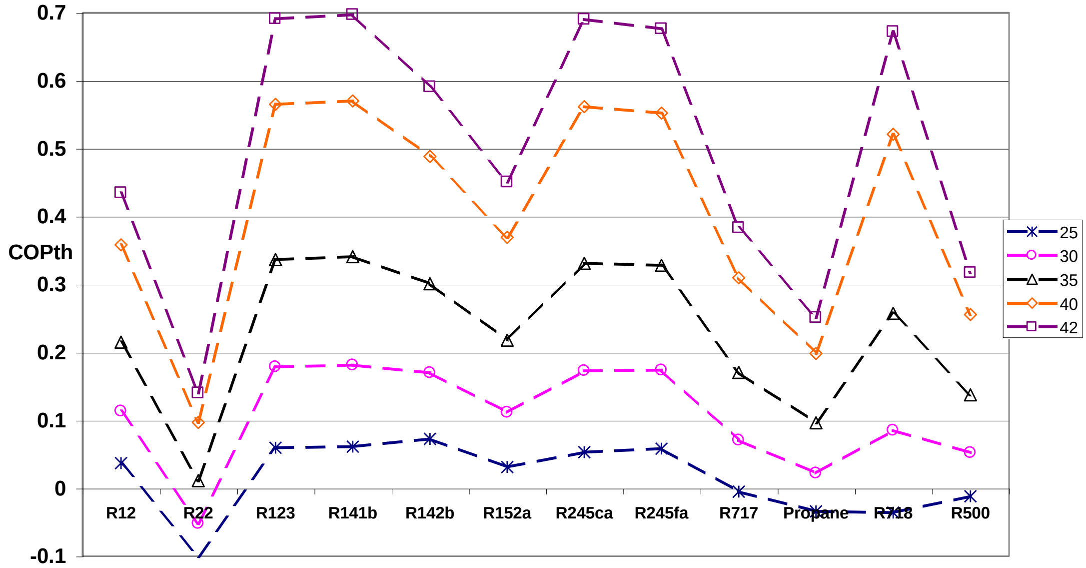

Figure 5. Responses of refrigerants towards changing intercooler temperature. COPth is averaged over combinations of refrigerant on ejector side (x-axis) with the rest, Tg = 80, Tc = 50, Te = 5). Legends show intercooler temperatures.

Shown in Figure 5 is a similar fashion of behaviour from the refrigerants. However, since the condenser temperature is increased to 50°C, the COPth resulted are markedly lower. In the event the generator temperature is increased to 90°C with fixed condenser and evaporator of 50 and 5°C respectively, all the refrigerants respond similarly towards increasing intercooler temperature. The only difference is that the COPth values of the refrigerants are now higher for the same intercooler temperature range (25-42°C).

5. CONCLUSIONS

The present study proposes several alternative refrigerants that have potential for long-term use with minimal environmental effects in the solar powered combined ejector vapour-compression system. Criteria of an ideal refrigerant for the combined system are described. It should be understood that it is hardly possible to find a perfect refrigerant that meets all those criteria. The physical & thermodynamic, environmental impact, cost and performance comparisons were carried out for the refrigerants considered. Performance indicators in terms of entrainment ratio and area ratio for the ejector sub-cycle, COP for the vapour compression sub-cycle and COPth & COPme for the combined system with the refrigerants as a single working fluid are tabulated.

The refrigerants were also simulated in pairs to find the best combinations that give the highest overall COP at varying intercooler temperature under two sets of generator and condenser temperatures. This is because a change in intercooler temperature has a strong effect on the overall system performance and invites different responses from different refrigerants of specific properties. While there is not one refrigerant pair that is absolutely best, R141b as a single refrigerant in the combined cycle dominates at most of the intercooler temperatures. From the proposed refrigerants, R245ca and R245fa on ejector side paired with R141b on vapour-compression side are very competitive in terms of performance. These two refrigerants have the right saturation curve for ejector expansion, low critical pressure, favourable characteristics in terms of environmental impact, physical and thermodynamic properties except that R245ca is flammable with rather high viscosity and price.

While simulation was carried out for varying intercooler temperature, it was observed that COPth and COPme have opposite trends with different extent in response to intercooler temperature change. COPth also responds differently to generator temperature change as opposed to COPme. These findings will enable us to determine an optimum intercooler temperature for a given operating condition in our future work.

Also noticed is the fact that water tends to perform gradually better at increasing intercooler temperature. This is largely due to its fast developing saturation pressure at increasing intercooler temperature with steady high latent heat of evaporation, which leads to higher entrainment rate of the flow across the evaporator and thus higher refrigerating effect. It is very desirable to select water/steam for use in the ejector side due to all its well-recognised advantages.

While ammonia as a single refrigerant may perform better than the rest of the proposed refrigerants in a set of operating conditions in the combined cycle, its COP value was still low (Table 1). This performance did not carry on when the intercooler temperature was increased in the pairing simulation. Moreover, even though ammonia is natural, its very high saturation pressure, flammability and toxicity counteract its good performance.

The sensible selection of refrigerant pair would seem to be R245fa (or water/steam at high intercooler temperature) on the ejector side and R141b on the vapour compression side.

Further study involving simulation of the ejector processes in axisymmetrical CFD model would give deeper insights and validation of the results given in this study.

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