ASSESSMENT OF LIFE CYCLE CLIMATE PERFORMANCE FOR CHILLERS

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The use of Life Cycle Climate Performance (LCCP) assessment is becoming a preferred alternative to the Total Equivalent Warming Impact (TEWI) calculation because it includes consideration of the equivalent warming impact of the refrigerant during production as well as during use. However some recent studies using LCCP methods have concluded that fluorocarbon-based chillers are more environmentally friendly than those which use natural refrigerants. This conclusion is examined in detail in this paper, and LCCP calculations are detailed for existing water cooled chillers using R-717 and R-134a, existing air-cooled chillers using R-717 and R-134a and a promising alternative system using R-744.

1. INTRODUCTION

In March 2002 a report was made to the Alliance for Responsible Atmospheric Policy by the Arthur D Little Consultancy of Cambridge, MA. This report's findings have been quite influential in steering the thinking of policy making bodies, possibly including the various government departments which advise the decision-makers in the United Kingdom. However some of the conclusions of the report are surprising and therefore prompt further scrutiny.

2. EFFICIENCY CALCULATION METHODS

In the early 1990s the concept of Total Equivalent Warming Impact (TEWI) was introduced to help to clarify the choices facing system specifiers and designers. TEWI assigns a single number rating to the system representing a combination of direct and indirect effects. The direct effect is the consequence of release of the refrigerant to atmosphere, and the indirect effect is the impact of carbon dioxide emissions from fossil fuels, and hence is a function of energy efficiency. Both of these elements require a subjective judgement of the operation of the plant to be made. The indirect effect is based on an estimate of the total energy consumption of the system per year and the direct effect is based on an estimate of the leakage rate. Despite these weaknesses, the attraction of a single-number rating resulted in widespread use of the method, and it was even included in an appendix to part 1 of EN-378:2000. Provided the system is only used for reasonable comparison between similar systems, and the same boundary conditions and operating preferences are applied, it provides a useful general indication; and as it tends to increase the focus on system efficiency it is a welcome move. To reduce the subjective element in this type of calculation the Air-conditioning and Refrigeration Institute in Arlington, VA (ARI) introduced the concept of an Integrated Part Load Value, in their standard 550/590 on chiller performance rating. IPLV is based on the recognition that chiller performance and external weather conditions are variable, and rather than attempting a complex simulation it assigns reasonable values to the distribution of chiller performance data. The reasonable values are based upon an analysis of a particular building in a specific location (Atlanta, GA), but they are assumed to be typical values, and, like TEWI, provided the same values are applied to all systems under consideration, a useful general comparison can be drawn.

A further refinement of the TEWI concept was the more recent introduction of Life Cycle Climate Performance indicators (LCCP). This calculation includes the two main elements of the TEWI calculation, but also includes an assessment of the environmental impact of refrigerant production, distribution and end-of-life disposal. This impact includes an estimation of the energy required to run the production process, as well as the effect of so-called "fugitive emissions"; losses to atmosphere of refrigerant and feedstock or byproducts. Natural refrigerants such as ammonia and carbon dioxide do not cause environmental harm through fugitive emissions – the majority of carbon dioxide supplied in solid or bottled gas form is recovered from the waste streams of industrial processes. However even these plants require some energy to operate. LCCP gives the impression of a more rigorous treatment than the TEWI method, but in practice the values assigned to these activities have a second-order, or even third-order significance in comparison to the energy efficiency of the system in operation, and they are subject to the same subjective estimates of factors on performance and leakage rates as the original TEWI concept. Some published data on LCCP values has been collated and is presented in Table 1

Chemical	GWP_{100} (kg CO_2eq/kg)	Fugitive emissions (kg CO_2eq/kg)	Embodied Energy (kg CO_2eq /kg)	Total (kg CO_2eq /kg)
R-22	1700	390	3	2093
R-134a	1300	4 (78)[1]	9	1313 (1387)[1]
R-404A	3260	36 (111)[2]	9	3305 (3380)[2]
R-717	nil	nil	2	2
R-744	1 (0)[3]	nil	1	2 (1)[3]

Table 1 – Summary of some Life Cycle Climate Performance Data

Notes: [1)]Most fluorocarbon data is from the ADL 2002 report, but the higher figure reported in brackets for fugitive emissions is from an AFEAS study.
[2)]The R-404A fugitive emissions figure is based on the ADL report with an additional allowance for losses during the blending process
[3)]It has suggested (for example by Lorentzen in 1994) that the GWP of industrial CO_2 which would otherwise have been vented to atmosphere should be taken as zero.

In seeking to interpret these figures it should be borne in mind that the accuracy of the GWP figures is approximately +/-20%, which is far in excess of all the other figures included in the total, even under the most pessimistic assessment of production figures, with the exception of R-22 fugitive emissions. This figure is significantly higher as a result of releases of R-23 during the production process.

3. COMPARISON OF PUBLISHED DATA

The 2001 edition of the ASHRAE Fundamentals Handbook contains in Table 8 of Chapter 19 a comparison of likely efficiencies of various refrigerants calculated over a number of operating conditions. This shows that the low pressure refrigerants such as R11 and R123 show excellent cycle efficiencies in air conditioning applications, primarily as a result of their very high critical temperatures. Section G of the table gives performance as kW power consumption per kW of cooling at suction and discharge conditions of 277K (+4^{o}C) and 310K (+37^{o}C) respectively, with no superheat. The three most efficient refrigerants in the table are R-11, R-113 and R-123, with Power Consumption Ratios of 0.133-0.135 kWe/kWr. The only other refrigerant offering an efficiency below 0.14 is ammonia, with an PCR of 0.137, and R-134a is listed as 0.144 (5.1% higher). Of all the medium and high pressure fluorocarbons in use as refrigerants over the last ten years, only R-22 has a better ratio (at 0.142) than R-134a.

In the Arthur D Little report of 2002, entitled "Global Comparative Analysis of HFC and Alternative Technologies for Refrigeration, Air Conditioning, Solvent, Foam, Aerosol Propellant and Fire Protection Applications", chapter 7 provides an overview of chiller performance, as viewed by the authors. This view gives quite the opposite impression to the ASHRAE table, suggesting that a Life Cycle Climate Performance calculation shows that fluorocarbon chillers for air-conditioning are generally likely to account for 10% lower carbon equivalent emissions than ammonia chillers, and suggesting that the energy consumption of R-134a and R-22 chillers is 14% lower than an equivalent sized unit with ammonia. In contrast to the ADL report, a paper presented by York International at the IIAR's annual meeting in 2003 provided detailed performance information for ammonia and R-134a chillers, contrasting "standard" chillers of each type with the "best" available on the market (Tychsen, 2003). This found that the standard ammonia chiller was over 9% more efficient than the standard R-134a model, and the best ammonia chiller was over 17% more efficient than the best R-134a model. This report used the same "Integrated Part Load Value" method as the ADL report, and it examines the same types of chiller (water cooled with screw compressors) but it reached significantly different conclusions. Part of the complexity in this field arises from the different methods of expressing efficiency data. For the purposes of this paper, the lead given by ASHRAE will be followed, with power consumption expressed as kW of energy consumed per kW of cooling. Where appropriate the inverse of this ratio, the coefficient of performance (CoP) will also be given. A summary of the figures from these three sources is given in Table 2.

Source	Ammonia PCR (CoP)	R-134a PCR (CoP)
ASHRAE 2001 Fundamentals Ch 19 table 8.G	0.137 (7.30)	0.144 (6.94)
ADL Report 2002, ch 7 "ideal cycle"	0.150 (6.66)	0.159 (6.27)
ADL Report 2002, ch 7 1200kW chiller IPLV	0.154 (6.51)	0.142 (7.03)
ADL Report 2002, ch 7 3500kW chiller IPLV	0.162 (6.17)	0.136 (7.33)
Tychsen – IIAR 2003, 1100kW best chiller IPLV	0.121 (8.27)	0.142 (7.05)
Tychsen – IIAR 2003, 1100kW standard chiller IPLV	0.134 (7.47)	0.146 (6.83)

Table 2 – Comparison of performance data from various sources

The Tychsen paper gives clear details of the sources of the figures used, but the ADL report is less transparent. There appears to be a conflict of assumptions in the ADL calculations, as it is evident that the use of IPLV calculations improves the calculated energy efficiency of the R-134a chiller but has the opposite effect for the ammonia chiller. This dichotomy is not seen in the Tychsen calculations, and it is not consistent with operating experience of chillers of this type. One possible explanation of this anomaly is that the ammonia chiller data, which is not detailed or referenced in the ADL report, may be based upon a system with a fixed head pressure, whereas the R-134a systems in the report can reduce the discharge pressure in times of low load and/or low ambient. If so, then this is rather at odds with typical installations, where the ammonia plant, which is less likely to use a traditional thermostatic expansion valve, is more likely to be configured to gain the benefit of floating head pressure.

4. LARGE AMMONIA CHILLERS – A FRESH ASSESSMENT

As the ADL report gave a very pessimistic view of large ammonia chillers, and units of this size were not considered in the Tychsen paper, it was decided to complete an IPLV calculation for a unit with a single large screw compressor and a nominal capacity of 3,500kW. It is assumed that this unit uses plate heat exchangers as the evaporator and condenser, and that it is controlled by a high pressure float system, allowing the head pressure to float. Several systems of this type, with capacities ranging from 1,000kW to 3,500kW have been installed in air-conditioning applications in the UK over the past five years, so there is a reasonable spread of case history to compare against the calculation. An industrial twin screw compressor with a swept volume of 3,250 m^3/h and capacity of 3,760kW when running economised with a suction of 4.4°C and discharge of 40°C was used for the simulation. It should be noted that this was a test calculation, and was not based upon a real chiller, so the calculated values could not be compared with a specific system. However the calculation strictly followed the ARI 550/590 IPLV method. This showed a Power Consumption Ratio of 0.182 (a CoP of 5.47) at the design condition of 3,500kW and 25°C wet bulb temperature. As the ambient and load reduce, the PCR can be substantially reduced, with a minimum of less than 0.1 kW/kW when the ambient is less than 10°C (wet bulb) and the load is between 50% and 90% of design. The four rating points for the IPLV calculation are loads of 100%, 75%, 50% and 25% in ambient wet bulbs of 25°C, 19.4°C, and 13.9°C for the lowest two loads. These temperatures follow the ARI guidance of entering condenser water temperatures of 85°F, 75°F and 65°F with a 8°F approach.

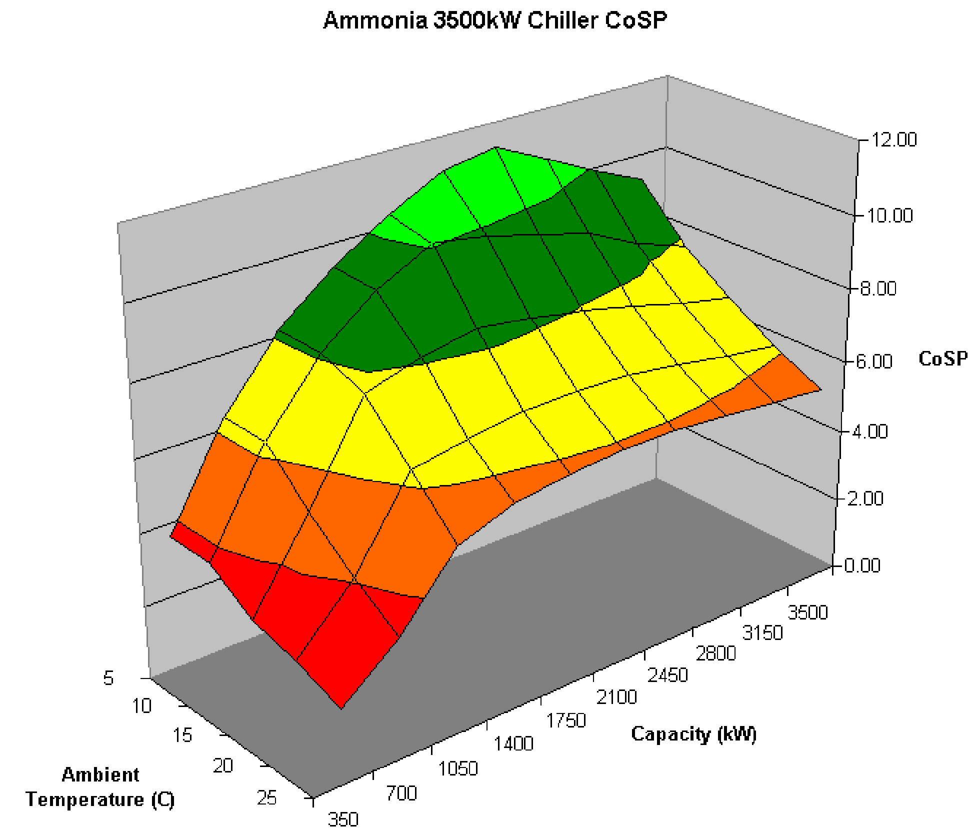

Figure 1: Performance map for large ammonia chiller – single screw compressor

These parameters result in an integrated part load value of 0.132kWe/kWr, equivalent to a CoP of 7.6 which compares well with Tychsen's figure of 0.134 for a standard ammonia chiller. A map of CoP contours for this range of loads and ambients is shown in Figure 1. The exercise was repeated with data for a similar compressor on R-134a, this time sized to provide approximately 4,000kW with the same design conditions. The evaporator in this case is assumed to be a flooded shell-and-tube with enhanced copper tubes giving an approach of about 2K. The compressor, also an industrial twin screw, has a swept volume of 5,800 m^3/h and capacity of 3,950kW when running economised with a suction of 4.4°C and discharge of 40°C. This gives an integrated part load value of 0.169kWe/kWr, equivalent to a CoP of 5.9 which compares less favourably with either the Tychsen figure of 0.146 for a standard R-134a chiller or the ADL figure of 0.136. The map of CoP contours for this range of loads and ambients is shown in Figure 2.

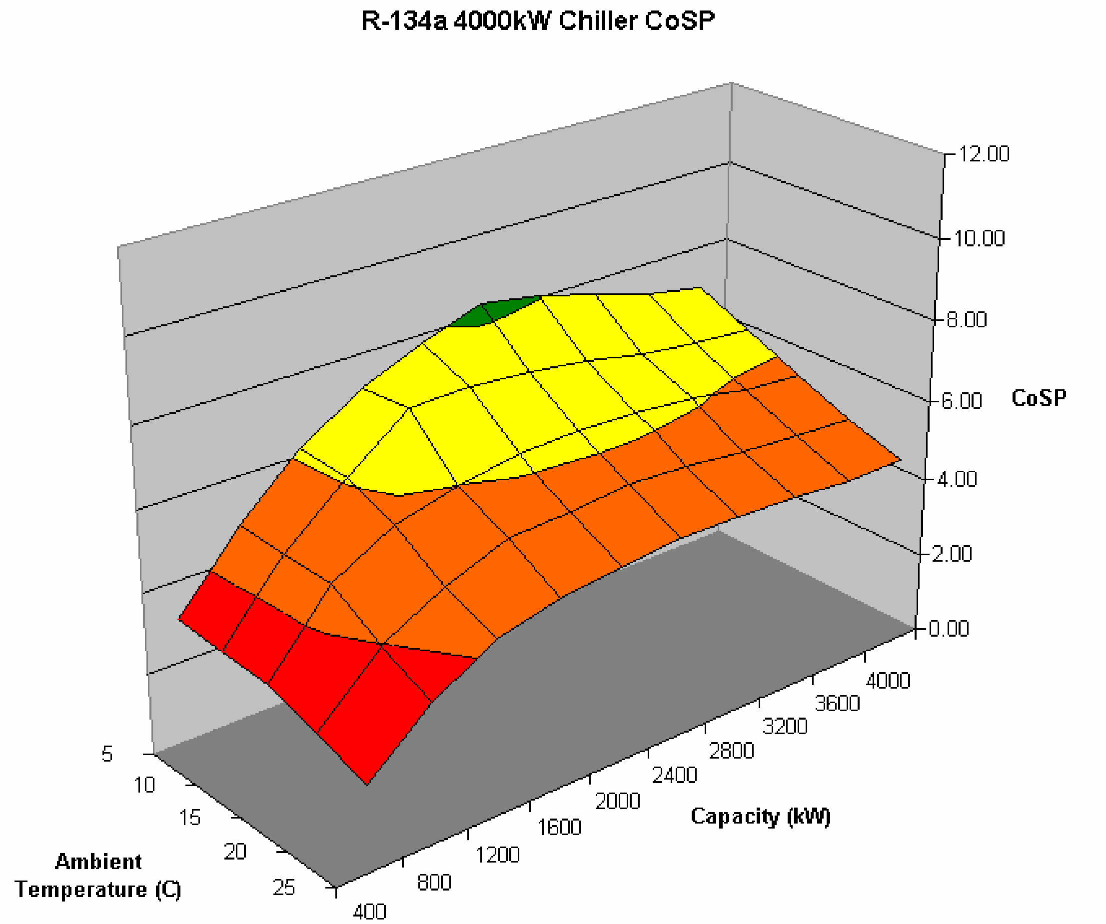

Figure 2: Performance map for large R-134a chiller – single screw compressor

Comparison shows three reasons why the ammonia IPLV is significantly better than R-134a. The full load CoP is 5.47 for ammonia and 4.73 for R-134a, so the whole surface is at a higher level for ammonia. As the ambient reduces the gradient of the ammonia surface is steeper, suggesting that the compressor is more able to take advantage of lower head pressures in cold weather. The same gradient can be observed for part load operation in colder ambients, resulting in a peak CoP for ammonia of more than 11, compared with just over 8 for R-134a. Two possible explanations for the difference observed at lower condensing pressures are that ammonia has a higher index of compression than R-134a (typically 1.4 compared with 1.18) and that it has a much higher critical temperature. Both systems were configured to benefit from variable volume ratio, and both were assumed to have an economiser: these features would normally be expected to be of greater benefit to the R-134a system. It should however be noted that the minimum volume ratio is 2.6 on this type of screw compressor, and that this translates to an “internal” pressure ratio of 3.81 for ammonia and 3.03 for R-134a. At the design condition of 4.4^{o}C suction and 40^{o}C discharge their actual pressure ratios are 3.08 and 2.97 respectively. This means that they are both overcompressing across the entire operating range of discharge pressure conditions. The compressors will run at all times on the minimum volume ratio, so there is no benefit from the variable ratio function. The economiser is of limited benefit because the saturated suction temperature is relatively high, so as the discharge condition drops, the amount of subcooling which can be provided by the economiser diminishes, until with a discharge pressure of 20^{o}C it is negligible. One side effect of the low critical temperature of R-134a is that good performance is dependent on having an effective economiser, so during low head operation the adverse effect is more obvious.

5. MEDIUM SIZED, AIR COOLED CHILLERS

Many smaller chillers use air cooled condensers rather than cooling towers, so the modelling exercise was repeated for two typical types of smaller unit. The ammonia unit has three screw compressors, with capacity slide valves, and has a total capacity of 700kW condensing at 55 ^{o}C in an ambient of 35 ^{o}C. The IPLV is 0.22kWe/kWr; an integrated CoP of 4.57. The CoP contour map is shown in Figure 3.

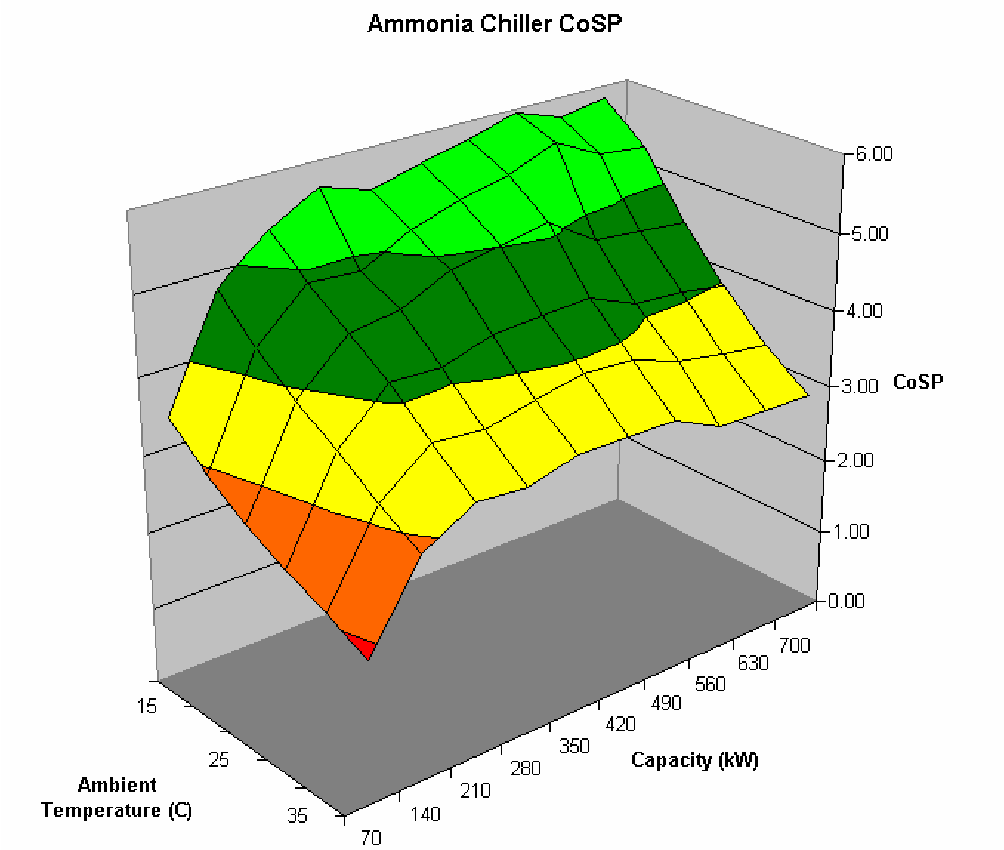

Figure 3: Performance map for small ammonia chiller – three screw compressors

The control algorithm assumed for this pack was that all three compressors offload together until one is no longer needed, when it is switched off and two run at full load. It can be seen that the CoP is consistent across most of the envelope, but falls sharply when the load drops below 20%. It can also be seen that the benefit of floating head pressure in cold weather is significant.

The alternative to this using R-134a was modelled on a rack system, providing 600kW at the same design point, but using eight semihermetic reciprocating compressors. Capacity control is by switching off compressors, so in theory the part load CoP should be very good. In practice however the increased compressor capacity achieved by running at lower discharge pressure is not sufficient to offload more than one compressor at full load as the ambient falls (the compressor capacity rises from 73kW at 55^{o}C condensing to 92kW at 35^{o}C condensing). The map is shown in figure 4, and displays the same flatter characteristic as the R-134a screw compressor map in figure 2.

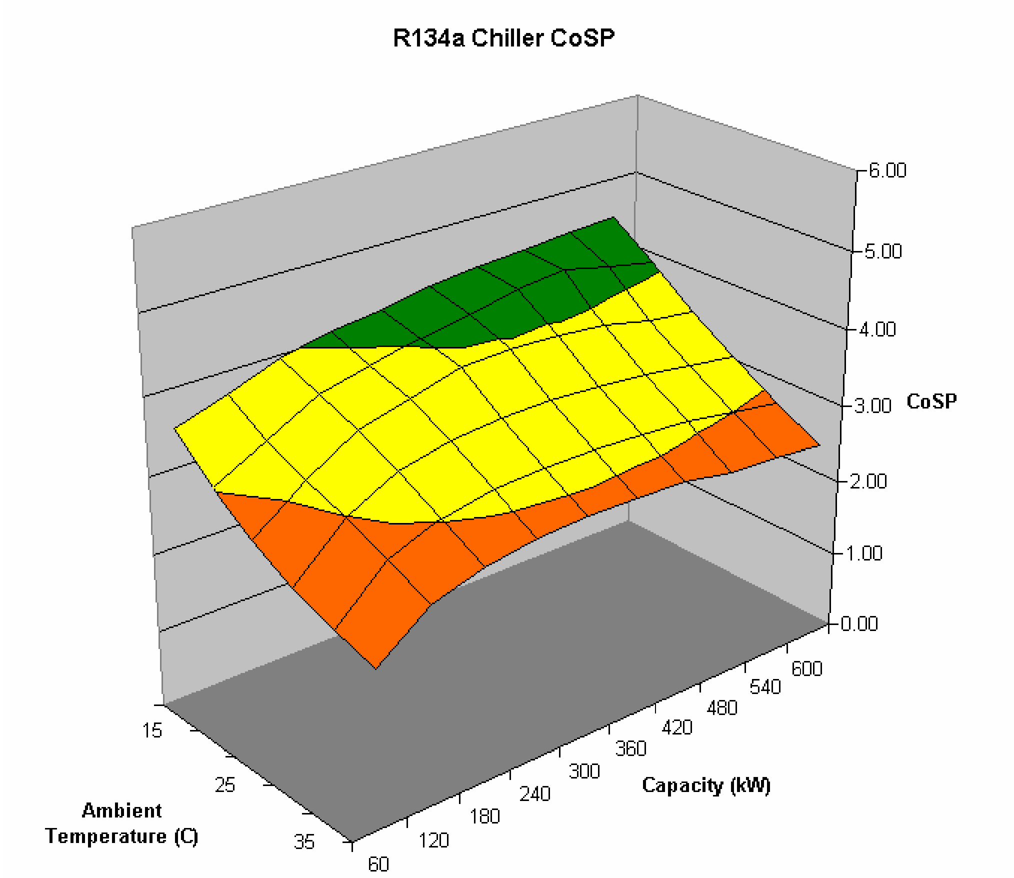

Figure 4: Performance map for small R-134a chiller – eight recip compressors

The integrated part load value for this chiller is 0.27kWe/kWr, which is equivalent to an integrated CoP of 3.66. It should be noted that the power consumption of the chillers illustrated in figures 3 and 4 include the air cooled condenser fans, which are assumed to run at full speed until the low head pressure limit is reached, at which point they are stepped down in sequence. The benefit of eight small compressors at the minimum load condition can be observed in figure 4, where the CoP does not fall as sharply as for the three ammonia compressors. However the dip which is observed here is the effect of the condenser fans, which consume nearly as much power as the single compressor when the chiller is on minimum load.

6. FUTURE DEVELOPMENTS

Both ADL and Tychsen make the observation that the efficiency of the chillers in question can be enhanced by the use of variable speed drives. This is certainly true for centrifugal compressors, for twin screws and for reciprocating machines, but the following constraints should be observed. Centrifugal compressors depend upon the rotational speed to develop the discharge pressure, so speed variation is a good way of exploiting low ambients to improve efficiency, but it is not so appropriate for low loads when a high head is still required, as the compressor will tend to surge. In this case suction throttling is necessary, and it is not really efficient, even if inlet guide vanes are used. For screw compressors the optimum efficiency is related to rotor tip speed, with a 5:1 turndown ratio between maximum efficient speed and minimum efficient speed. Once the speed drops below this minimum, the volumetric efficiency drops sharply as the slip increases. Compressors, with small rotor diameters have a high maximum – up to 8,700rpm for 110mm – but consequently their minimum efficient speed is also high – in this case 1,750rpm. This means that using an inverter to slow down a 2 pole motor on a small screw compressor is not really practical because the minimum speed is 60% of the direct on line speed. Of course for small compressors an inverter can be used to run to higher speeds, thus producing a more useful turndown ratio, but in this case the electrical losses in the inverter, typically about 3% of full load, must be included in the overall calculation as the inverter cannot be bypassed. With reciprocating compressors the limit on minimum speed is usually the type of oil distribution used. With a gear pump driven from the crankshaft the minimum speed is quite high – typically 40% of the maximum, so if low speed is required a splash lubricated machine should be used. This is usually only found in smaller compressors. A further factor to bear in mind is that frequency inverters for speed control of larger chillers are still not commonly available. The large chiller modelled in figure 1 would require a 700kW drive motor.

A further development in the context of water chilling for building services is the concept of using carbon dioxide instead of water for the distribution of cooling. In its simplest form, such a cooling system could use an R-134a package to condense carbon dioxide for circulation to evaporators in air handling units. It should be noted that the IPLVs quoted for all types of chiller in Table 1 did not include the power required for chilled water pumps, as these were assumed to be similar for the systems under comparison. As these pumps run continuously the energy consumed is significant, and can account for as much as the compressor motors. Use of carbon dioxide would present significant design challenges, but the pump power would be reduced by a factor of about 10, and so the power consumption ratio for an air-cooled system, which might be 0.5kWe/kWr, could be reduced to about 0.3kWe/kWr <u>including distribution pumps</u>. Clearly the practice of considering compressor power and ignoring distribution power cannot continue. Even greater benefit could be gained by using high pressure carbon dioxide compressors to reject heat at transcritical conditions. A design concept was developed to investigate this possibility. This design assumed that carbon dioxide was circulated to air handling units at an evaporating temperature of 10^{o}C, instead of using water at 6^{o}C flow and 12^{o}C return. In ambient temperatures above 20^{o}C dry bulb the suction gas from the AHU coil is compressed to 85 Bar(A), and this discharge is cooled in counterflow heat exchange with a low volume airstream. The power required for the heat rejection is negligible because the air flow is so small in comparison with a traditional air-cooled condenser; the gas cooler for this system would heat air to about 90^{o}C, and would be more like a boiler flue than a normal condenser. In lower ambients a bypass expansion valve allows the discharge pressure to float at sub-critical conditions, with a minimum condensing condition of 20^{o}C. The estimated performance of a transcritical carbon dioxide system as described is shown in figure 5. The simple model showed even higher performance figures, but for the purposes of this comparison the CoP has been capped at 6 under all conditions.

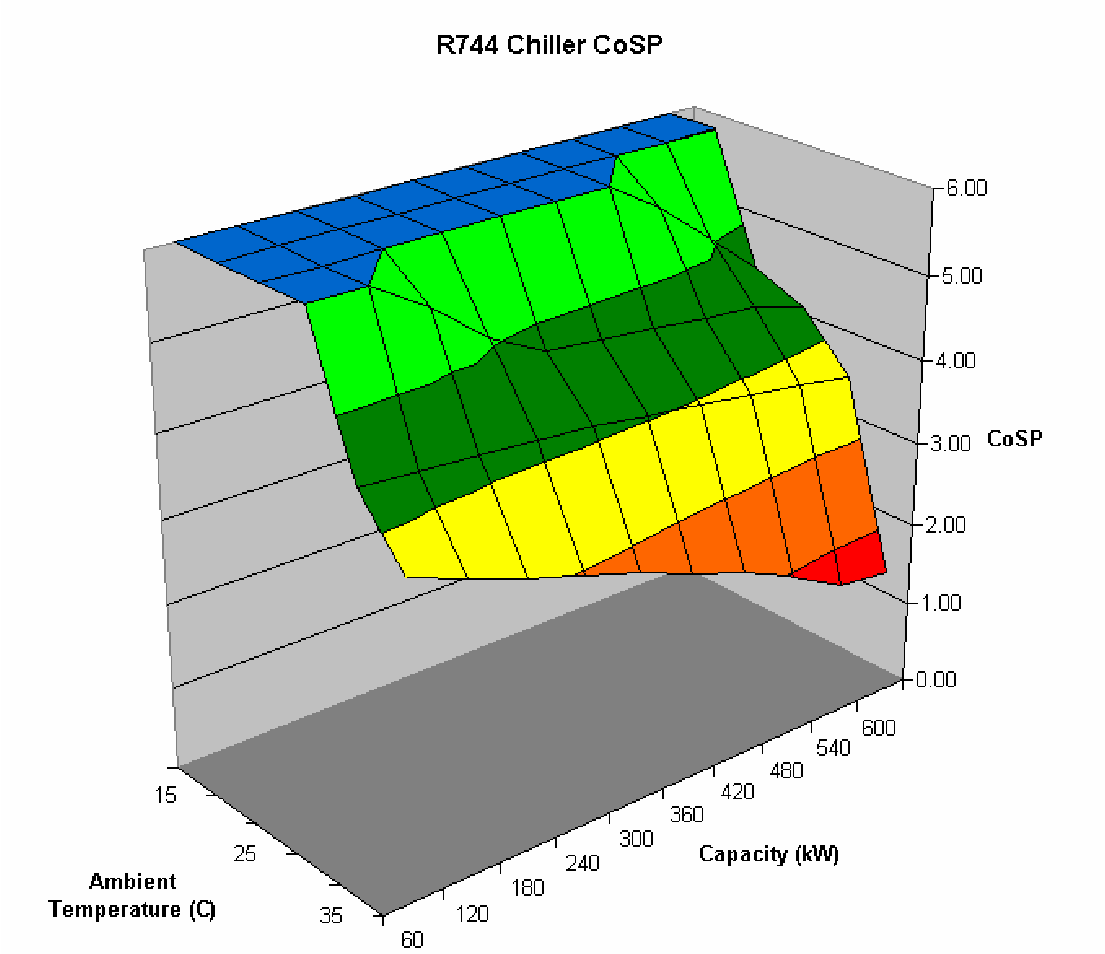

Figure 5: Estimated Performance map for carbon dioxide central system (note CoP capped at 6.0 for this estimate)

The integrated part load value for this chiller is 0.19kWe/kWr, which is equivalent to an integrated CoP of 5.26. It should be noted that the power consumption in this model does not include carbon dioxide pumping power. If this is estimated on the same basis as before then the total power requirement (including pumps) for the transcritical carbon dioxide system would be 0.22kWe/kWr, compared with 0.30kWe/kWr for the secondary carbon dioxide system (with a R-134a high side), 0.47kWe/kWr for air-cooled water chillers using ammonia and 0.52kWe/kWr for air-cooled water chillers using R-134a. The savings made by changing the heat transfer medium are far greater than the savings made by switching refrigerants, but the most efficient system is achieved by combining the best elements of both primary and secondary circuits.

7. CONCLUSIONS

The suggestion in the Arthur D Little report that R-134a chillers are inherently more efficient than equivalent ammonia units does not bear close examination. On the contrary the ammonia chillers under typical operating conditions are likely to be about 20% more efficient than the R-134a equivalent using an IPLV method. This efficiency benefit swamps all other considerations under TEWI and LCCP. Migration to new technologies will offer further energy benefits to both families of "chiller". In the case of variable speed drives the benefit will be incremental, in the case of a shift to carbon dioxide in place of chilled water the benefit will be substantial. In the near future, secondary carbon dioxide systems could be implemented relatively easily, but additional development work is required before the full benefit of a transcritical heat rejection system can be gained. It is essential to focus on system efficiency issues in order to gain a true perspective on the environmental impact of chiller systems.

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