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CHALLENGES IN DEVELOPING ENVIRONMENTALLY SAFE HEAT PUMPING SYSTEMS*

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ABSTRACT

The advent of the global warming crisis has brought about a viewpoint, in many governments, that the halogen family of refrigerants should be replaced, in part or in toto, by the so-called natural refrigerants. For this proposal to be valid it is necessary to consider both the refrigerant's direct environmental impact and its life-long performance under field conditions. This is particularly true in the case of global warming because, for most applications, it is the heat pump's operating efficiency and its impact on the central power plant's emissions that is the dominating environmental factor. Any refrigerant must also meet a variety of other criteria that deal with durability, safety and costs. A simple comparison of basic fluid properties is conducted to indicate what system design considerations must be made if a refrigerant is to become an acceptable alternative. It is also reasoned that while computer models and laboratory prototypes are a necessary beginning, they are not sufficient to determine the true environmental impact of any system. Finally, the question is raised as to whether the refrigerant specification approach is the better path to an improved environment or if it is wiser to leave all options open for researchers and manufacturers to meet an environmental performance standard any way they choose.

INTRODUCTION

If the refrigeration industry wants to meet its responsibility towards solving the global warming crisis, it must first consider both the direct impact working fluids have on the atmosphere and the indirect impact a given heat pumping system will have on the electric power plant emission rate. Considering the fact that the intent of any design of a heat pump is to retain the refrigerant within the system throughout its lifetime, it should be of no surprise that it is the indirect effect that is the dominant issue. Numerous studies have shown that for almost every application of refrigeration, air conditioning, heat pumping, water chilling, etc.; the power plant's carbon dioxide (CO_2) emission which is expended over the lifetime of the refrigerant system far exceeds any impact from the refrigerant itself (Fairchild et al., 1991). These studies use indices of common measure to provide a single value of the working fluid in its application over the projected lifetime of the system.

The industry has been using the Total Equivalent Warming Impact (TEWI). This index is a composite of the direct environmental warming effect and the indirect warming effect. The direct effect is due to the release of the refrigerant into the atmosphere. The indirect effect is from combustion emissions generated by the central power plant used to power the refrigeration system. The consequence of these studies shows that it is essential that the field performance of any newly proposed concept be the paramount consideration.

The performance of a heat pump is strongly dependent on the working fluid's effectiveness, which is a function of its thermophysical properties, and on the type and quality of the hardware components that make up the system. Often the less effective working fluid's performance can be enhanced by the implementation of an additional or improved hardware component. For example, the replacement of CFC-11 with HFC-134a, in centrifugal water chillers, has caused at least one manufacturer to use an expansion turbine for work recovery, to partially recover the overall system performance loss caused by the refrigerant change. While this new system ensures the manufacturer's new HFC product line performance range will overlap his previous CFC product line performance range, it is not the most efficient system possible. A HCFC-123 two-stage centrifugal water chiller with an economizer cycle is more efficient and always can be since HCFC-123, itself, is more efficient than HFC-134a and equivalent hardware performance improvements can be made for either one. This latter point is important because there are cases where hardware improvements can be made for one refrigerant but not for the other. For example, ammonia has a significantly higher thermal conductivity than halogens making it a better heat transfer fluid. However, in an actual heat exchanger, where halogens can use enhanced copper surfaces and ammonia is restricted to steel, the halogen is the preferred choice. Thus, both the refrigerant and the system must be considered in order to arrive at the minimum atmospheric damage.

*Heat pump systems, as used in this paper, is meant in the physics sense, that is, it refers to any or all air conditioning, water chilling, refrigeration, or heat pumping systems.

The HFC-134a solution was perfectly satisfactory for the ozone crisis because the manufacturers could offer customers the traditional performance range (perhaps even an improvement) with a chlorine-free system. However, for the global warming crisis where the real culprit is system performance, it is the scheduled phase-out HCFC-123 system that is the best known solution for water chillers. This paradox comes about because the refrigerants were considered in isolation from the heat pumping system's field performance. Because the natural fluids cover a much wider range of vapor pressures and physical properties than any alternative fluid previously considered, it is expected that a wider variation in hardware systems will exist. In fact, two of the natural fluids, air and carbon dioxide, do not even operate on the traditional vapor compression cycle.

THERMODYNAMICS & REFRIGERANT PROPERTIES

The Carnot Cycle acts as the ideal model for a heat pump as it does for power generation. Its two isothermal and two isentropic processes result in a cycle efficiency that cannot be exceeded; albeit, it can be equaled. It represents a performance measure independent of the working fluid. The fundamental premise of all heat pumping systems is to transfer heat from a lower temperature source to a higher temperature sink by raising the pressure of the working fluid. The desire, then, is to move a maximum amount of heat with a minimum amount of fluid pumping. It is difficult to imagine a better way to accomplish this than by utilizing fluid phase change; both because of the thermal capacity per unit mass and because of the high heat transfer coefficients. Therefore, the two-phase Carnot Cycle with dry compression (denoted by Area 1-2-3-4) is illustrated in Fig. 1a as the maximum performance limit of a practical heat pumping system. T_L, T_H, T_e, and T_c are the heat source, heat sink, evaporator, and condenser temperatures, respectively. In the case of the Carnot Cycle, T_L and T_e and T_H and T_c are infinitesimally different from one another. Note that the areas on any temperature-entropy (T-s) diagram represents energy and the specific area, within the cycle boundaries, represents the required work. One practical deviation from the ideal Carnot Cycle is to introduce irreversible heat transfer processes due to finite temperature differences between the refrigerant in the evaporator and condenser and source and sink heat transfer fluids. These temperature differences are denoted by ΔT_L and ΔT_H, respectively. It is seen, then, that irreversible heat transfer processes have a strong influence on the magnitude of the work (represented by Area 1'-2'-3'-4') and thus the efficiency. This can also be seen by considering a second-law efficiency defined for a heat pump as:

$$\varepsilon_{HP} = \frac{\phi_{Q,H}}{W_{act}} \qquad (1a)$$

and for a refrigerator (or air-conditioner) as:

$$\varepsilon_{AC} = -\frac{\phi_{Q,L}}{W_{act}} \qquad (1b)$$

where $\phi_{Q,H}$, $\phi_{Q,L}$, and W are the availabilities associated with the high-temperature heat transfer process, the low-temperature heat transfer process, and the compressor work, respectively. After some manipulation, eqs. (1a)-(1b) can be rewritten for a heat pump as:

$$\varepsilon_{HP} = \frac{1 - T_L / T_H}{1 - T_e / T_c} \qquad (2a)$$

And for a refrigerator (or air-conditioner) as:

$$\varepsilon_{AC} = \frac{T_H / T_L - 1}{T_c / T_e - 1} \qquad (2b)$$

Plots of eqs. (2a) and (2b) are shown on Fig. 1b for T_L = 0°C and T_H = 40°C. It is also tacitly assumed that the work of the isentropic expansion process is reversibly transferred to reduce the required isentropic compression input work. This implies that these component irreversibilities compound to reduce overall heat pump efficiency. This is particularly true for machines where the expansion work approaches a magnitude similar to that of the compression work.

A simple expression for the heat exchange processes is given by:

$$Q = UA\,F\,LMTD \qquad (3)$$

where U is the overall heat transfer coefficient, which is a function of the respective fluids' convection coefficients. The log-mean-temperature-difference (LMTD) represents the temperature difference between the refrigerant and the external heat transfer fluid, which is usually air or a water solution, and F is an adjustment factor for the two fluids' relative flow geometry complexities. The point here is that there is a general tendency in the industry towards physically smaller systems, for the same thermal capacity. This is being done in order to "accommodate an expected doubling to tripling of the urban system in a habitable, efficient, and environmentally friendly manner" (Kates et al., 1999). Since heat exchangers are usually the largest components in a heat pump system, it is they that are being reduced. From eq. (3) it can be seen that if A is reduced and LMTD is not, for efficiency and thus for the sake of the environment, then increasing U, and to some degree F, is necessary.

The advantage of the reverse ideal Rankine cycle (aka, the ideal vapor compression cycle) is that three of the four processes of the Carnot cycle are nearly the same. Fig. 2 shows that, except for a small de-superheating portion of the condenser, the ideal vapor compression cycle's isobaric heat exchange processes are also isothermal, and under the two-phase vapor dome. The compression process follows an isentropic path, though it is somewhat longer in order to reach the saturated condenser temperature in the superheated region. This latter effect is inversely proportional to the molar heat capacity of the refrigerant and is generally minimal for halogens. Only the isenthalpic expansion process of the vapor compression cycle deviates completely from the Carnot's isentropic one. The penalty for this totally irreversible process is both a loss of refrigerating effect (capacity) and expansion work potential (see Fig. 2). The magnitude of these losses are identical in absolute value and vary as a function of the particular refrigerant's critical point and molar heat capacity (Didion, 1999). As a result most refrigerants selected for a given application operate in the 0.6 to 0.8 reduced condenser temperature range. With the tendency towards equipment size reduction, the fluid thermal capacitance, eqs. (4a) and (4b), indicate that the higher pressure, and thus higher density (ρ), refrigerants offer a lower system volume (V) advantage. For single phase fluids:

$$Q = mc_p\left(T_{out} - T_{in}\right) \qquad (4a)$$

and for two-phase fluids:

$$Q = m\,l \qquad (4b)$$

where $m = \rho\,V$, and c_p and l are the respective heat capacity and latent heat terms.

Fig. 3 is a graph of vapor pressure lines of four of the five natural fluids and four of the more popular halogens in current use. The scales of the coordinates are the logarithm of pressure and the negative inverse of temperature; thus it is a graphical version of the Clausius-Clapeyron equation. It has the advantage of illustrating the vapor pressure in a linear form to make easy the relative comparisons of fundamental thermodynamic properties that are crucial in a given fluid's potential as a refrigerant in the ideal vapor compression cycle. Air is not included in the figure because its critical point is too low to be utilized in a vapor compression cycle. It is used in an all gas cycle and its properties will be discussed later. The other four natural refrigerants are shown, along with the four halogens they are touted to replace. The refrigerants whose critical points are further from the upper limit of the application condenser temperature are inherently more efficient. However, as noted in Fig. 2, they are also lower in volumetric capacity, which usually requires a larger machine. This trade-off is inherent for all fluids. This trade-off also implies that no one refrigerant can be optimal for all applications (McLinden, 1988). The use of HCFC-123, for example, is limited to centrifugal compressor systems because of its very low density and hence its very large amount of vapor flow. Although its efficiency exceeds all others, the centrifugal compressor is applicable to larger water chillers, only. At the other end of the scale is the near-zeotropic mixture HFC-410A, the intended HFC replacement for HCFC-22. It is the highest pressure refrigerant currently being used in air conditioners and heat pumps. With a lower critical point temperature, its thermodynamic performance is poorer than HCFC-22; but one of its components, HFC-32, has rather superior heat transfer properties (i.e., low viscosity and high thermal conductivity). System tests of machines optimized for each of these refrigerants, have shown that the heat exchanger performance of the HFC-410A system can offset, via reduced mean temperature differences between the refrigerant and the external heat transfer fluid, its poorer thermodynamic performance. As a result, residential heat pumps utilizing each refrigerant have similar coefficients of performances (COPs).

The vapor pressures of these eight refrigerants are shown on the figure to provide some clues as to each fluid's fundamental thermodynamic performance within the vapor compression cycle. Also, as seen above, knowledge of the heat transfer properties is necessary to grasp the relative magnitude of the Carnot limitations when heat pump system performance is to be estimated. It should be recognized that, with the possible exception of the transcritical cycle, all of the natural fluids have been or are in use, in niche markets, throughout most of this past century. However, none have been able to displace halogens in the high volume markets where the applications are in close proximity to people. Whether this is because of economic, safety or performance reasons we have to consider carefully the consequences if any are to be compromised.

AMMONIA

Of the known refrigerants, ammonia is certainly one of the oldest and best understood. Its thermodynamic properties (e.g., high latent heat of vaporization, high critical point temperature, low liquid molar heat capacity) support its inherent high efficiency in the ideal vapor compression cycle. Years of use in a wide variety of applications have established a solid foundation for most system and application designs. On a performance basis, it is clearly competitive with most halogens. However, ammonia's limitations show up in the materials compatibility and safety areas.

The corrosive nature of ammonia with copper, zinc, and their alloys makes these materials unavailable for hardware components. One of the ramifications of this limitation is that it is not likely that ammonia will be adopted for small unitary systems. The performance and reliability of these systems depends heavily upon their hermetic compressor/motor systems, soldered pipe connections, and the enhanced heat transfer surfaces available in copper tubes (Starner, 1993). It is unlikely that a small ammonia system will be developed, so that it could compete either economically or performance-wise with halogen systems (Pillis, 1993).

In larger systems, ammonia becomes more performance competitive but still has limitations. Most safety codes would prevent the use of direct expansion (DX) / air coils in ammonia systems so air conditioning applications would be likely only where package water chillers make sense anyway. With plate heat exchangers becoming more cost effective and available in non-copper alloy materials, several packaged chillers are available in the 20 to 1200 kW range. For those halogen refrigerant systems that use a semi-hermetic motor it would be necessary to have motor shielding development to allow for ammonia cooling. Typically, these systems also have supplemental compressor cooling (except for oil flooded screws where cooling occurs within the compressor) because of the high discharge temperature inherent to ammonia's simple lightweight molecule. Supermarket DX/air systems pose a more difficult problem. An ammonia DX/air replacement would create an unacceptable safety situation. The introduction of a secondary heat transfer fluid would cause a significant decrease in system performance, due to the additional irreversible heat transfer processes (see Figs. 1a and 1b) in the refrigerant loop. Assuming only a 5 ^{0}C temperature difference for the average heat exchange, the relative Carnot efficiency would drop to 0.79. In addition, the total food case design would have to be changed because of the necessary pipe diameter increase required if the secondary fluid were single phase.

In large field erected refrigeration systems ammonia has always been widely used and, with the eventual elimination of its main competitor, HCFC-22, there is little reason to think that this will not continue. In fact, the current ammonia usage in this area may well expand. In these systems, hermetic motor/compressors and copper tubes and heat exchangers are not used anyway. Also, the large charge requirements make ammonia costs attractive. Thus, as Pillis states, "Ammonia is already the refrigerant of choice in large refrigeration applications for the previously-mentioned reasons. The only limitations to its use are code restrictions that become too costly to meet".

These code restrictions are concerned primarily with ammonia's toxicity and flammability. Innumerable documents have been written on this subject and the degree to which restrictions apply are closely aligned with application. For marine applications alone, the requirements are listed in the Lloyd's Register document "Guidance Notes for Ammonia Marine Plant". It has been stated that the objective of these notes is prevention of leakage and the containment and safe disposal of ammonia from the plant and that ammonia will only be accepted in indirect refrigeration systems incorporating a secondary refrigerant, such as brine (Stera, 1993). This, of course, will limit the size of a single ship ammonia plant to that which can be packaged. In general, the approach to replacing a halogen system with an ammonia one is to isolate the refrigeration system to a well vented space and to use a secondary heat transfer fluid to communicate to the space to be cooled. The light molecular weight of ammonia allows for open air venting (Stoecker, 1989); although for large quantities water pool vent is preferred, if available. The use of the secondary fluid entails the inevitable system performance penalty due to the extra heat exchanger irreversibilities.

HYDROCARBONS

The use of hydrocarbons, and in particular propane, as an alternative to halogens has similar advantages and disadvantages to that of ammonia. That is, they have similar performance characteristics as halogens, but significant safety concerns. In fact these concerns motivated the development of the halogen refrigerants to begin with; when, in 1928 Thomas Midgley was asked if he could develop a nontoxic, nonflammable refrigerant. What resulted were the halogens. It is irony indeed that we must now consider the mistakes of the past as a passable solution. Fig. 4 is an illustration of the degrees of flammability as decreed by ASHRAE. The abscissa may be loosely interpreted as easiness to ignite and the coordinate as energy released upon ignition. It is seen that while both ammonia and hydrocarbons (all are clustered close to the propane point) are flammable, they are of very different magnitude as far as a threat to safety. Furthermore, unlike ammonia, the more popular hydrocarbon refrigerants are heavier than air and thus, in some machine rooms, leaks can accumulate and pose the threat of explosion. Although the halogens shown are components of some of the current mixtures in use, they are used only in such quantities that the mixture is not flammable even under extreme leak conditions. All other halogens in current use are non-flammable (Class 1). In Europe, home refrigerators containing hydrocarbon refrigerants, of the order of 30 grams, are currently in use. There is discussion of extending this practice to residential heat pumps, which, in the USA, could require up to four of kilograms or so. European Standard EN378, Refrigerating Systems and Heat Pumps, classifies refrigerants according to their potential hazards. For each refrigerant, a maximum limit is specified, with the maximum allowable charge calculated by internal volume of the room multiplied by 20% of the lower flammability limit. For propane in a room of 4 m x 4 m x 3 m, the allowable charge would be 0.38 kg. There is also a maximum charge for systems depending on the occupancy of the area. The lowest maximum charge is for areas such as public places and restaurants, and the amount is 1.5 kg, also being subject to the above calculation procedure. The standard forbids the location of any ignition sources in the vicinity of systems containing flammable refrigerants. This presents a challenging situation for restaurants due to many electrical switches and hot surfaces. There is a clause in a draft ISO standard that does not permit the use of flammables in split system heat pumps because they can contain resistance heaters as back-up, being an ignition source. Thus, for the USA where heating and cooling heat pumps prevail, flammable fluids will be forbidden for all residential units. A few years ago one USA major manufacturer developed a residential propane heat pump, which performed as well as their HCFC-22 machine. They then requested Underwriters Laboratory (UL) to recommend the additional safety add-ons that would be required for their approval. The requirements added 30%+ to the cost of the unit. Other manufacturers' studies have yielded similar results.

Europe and the USA have a dichotomy of views on this issue, largely because of fundamental societal differences. There is no doubt, that a residential hydrocarbon heat pump or air conditioner can be made to operate in a 'safe' manner. Numerous risk analyses show acceptable failures. However these analyses seldom consider the risks during repair, which is when virtually all accidents occur. There is also the issue of the manufacturer having to store and handle thousands of kilograms of highly flammable fluids. (In the last 30 years, the three biggest manufacturers have each had a major factory explosion from their "nonflammable" halogen tanks.) However, the most significant issues in the USA are public perception and the litigious society that the USA has become. In fact, it isn't a matter of whether the heat pump actually caused a fire; but rather that every fire offers the heat pump manufacturer as a potential 'rich' organization to pursue through litigation. It becomes a business liability decision that heat pump manufacturers must face. In order to adopt flammable fluids for this residential product line (a line that produces in excess of 6,600,000 units annually), a torturous path of consensus standards development and building code modifications must begin, prior to redesign of the products.

CARBON DIOXIDE

Carbon dioxide (CO_2) is one of the oldest and safest refrigerants known. It had been used in vapor compression cycle machines into the first half of the twentieth century. Its inherent limitation, as a refrigerant, is its low critical point temperature (see Fig. 3). The practical ramifications of this limitation are low efficiency and applicability to situations where the condenser temperature is below ambient. In response to the global warming crisis, Professor Lorentzen, of Norway, proposed a CO_2 transcritical cycle where the refrigerant rejected heat while it is in a supercritical state (see Fig. 5). The condenser is replaced by a gas cooler within which the refrigerant vapor is cooled without going through a condensation stage. This basic concept certainly improves the CO_2 performance and expands the potential field of application to include those that ultimately reject heat to the outdoor environment. The price, however, is high operating pressures, as much as 100 bars for certain applications. Considering that typical pressure vessel design safety factors are on the order of at least three; unique compressors and gas coolers are being developed. Most probably the capacity of such systems would be economically limited to residential and small commercial sizes. Considerable research has been conducted over the past decade and numerous applications have been identified as potential CO_2 candidates (Pettersen, 1999). Although production compressors are not yet available, it has been touted that less work per unit capacity will be required because of the lower compression ratio associated with the high pressure operating range. Also due to the sensible temperature glide associated with the gas cooler, it has been noted with prototype heat exchangers that some performance benefits can be realized in this component as well.

While the efficiency of the transcritical cycle has been improved over the CO_2 vapor compression cycle, the current transcritical cycle still is not competitive with current halogen systems. This is due to excessive expansion side losses; whereas a typical vapor compression cycle machine may require 15% flash gas to return the refrigerant to evaporator conditions, the CO_2 transcritical cycle may require more than twice that amount. Studies have been done on expansion side losses of the vapor compression cycle because some of the HFCs had greater losses than the CFCs they were replacing. These studies can at least act as an indicator to possible solutions to the transcritical cycle. Domanski (1997) evaluated three possible remedies: liquid-line/suction-line heat exchanger, economizer cycle, isentropic expansion engine. They are listed here in order of their effectiveness and hardware complexity.

WATER

Water is perhaps the oldest and one of the safest refrigerants known. It has a high critical point temperature and, as can be seen from Fig. 2, will have a large COP when used in a vapor compression cycle. This, however, will be accompanied by low volumetric refrigeration capacity due to the extremely low vapor density at the evaporator exit. For example, at an evaporation temperature of 0°C, water's vapor density is 0.004852 kg/m^3 which is a little more that four orders of magnitude lower than the vapor density of HCFC-22 at the same evaporation temperature. Low vapor density implies the need to move very large quantities of refrigerant through the system. For example, at this same evaporation temperature, the volume of water vapor that would need to be circulated is nearly 275 times more than the amount that would be needed if HCFC-22 were used instead. To grasp the magnitude of the amount of water that would need to be pumped through a system, consider that at an evaporation temperature of 0°C nearly 0.9 m^3/s of water would need to be circulated in order to obtain 10 kW of cooling. The required volumetric flowrate, however, is a strong function of evaporation temperature. For example, at an evaporation temperature of 20°C, the required volumetric flowrate would be 0.26 m^3/s. These large volumetric flowrates also imply large diameter piping since pressure drop is proportional to velocity, which is inversely proportional to the square of the pipe diameter. In addition to low vapor density, the required compressor pressure ratio is high (approximately 20 for an evaporation temperature of 0°C and a condensation temperature of 50°C). These factors combine to make normal mechanical compression difficult in heat pumping applications. Water further suffers from the problem that it results in extremely low evaporation pressures (e.g., at an evaporation temperature of 0°C, the saturation pressure is 0.6 kPa), which makes it difficult to construct practical, cost-effective equipment which can operate under these conditions. While it is true that some other common refrigerants, e.g., CFC-11 and HCFC-123 also operate under vacuum conditions at this same evaporation temperature, their saturation pressures are considerably higher at 40 kPa and 33 kPa, respectively. Due to all of these limitations, the use of water is limited, and will likely remain so. It, however, is currently being used to a limited extent in some applications such as absorption refrigeration, high-temperature heat pumping, vacuum ice production, and very large water chilling applications operating as open systems with direct-contact heat transfer.

AIR

Air, like water, is harmless to the environment. To date, some applications using air as a refrigerant have been food freezing, low temperature refrigeration, and aircraft and train cooling, among others. The air cycle follows the Joule (Reversed Brayton) Cycle, and can be either open or closed. For example, a *T-s* diagram for a closed cycle with an additional internal heat exchanger, which improves thermal efficiency, is shown in Fig. 6. The theoretical COP of the basic Joule (Reversed Brayton) Cycle (1-2-4-5-1), consisting of two isentropic and two isobaric processes, is given by:

$$COP = \frac{1}{\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} - 1} \qquad (5)$$

and is always less than that of the Carnot Cycle. In eq. (5), k is the ratio of specific heats. To achieve a large COP, the pressure ratio should be as small as possible. In real systems with non-isentropic compression and expansion processes, the resulting COP will be lower than the one given by eq. (5), and is a more complex function of pressure ratio and isentropic efficiencies as given in eq. (6) (Henatsch and Kauffeld, 1992).

$$COP = \frac{\eta_e T_{41}\left(1 - P_{21}^{\frac{1-k}{k}}\right) - \left(T_{41} - 1\right)}{\eta_c^{-1}\left(P_{21}^{\frac{k-1}{k}} - 1\right) - \eta_e T_{41}\left(1 - P_{21}^{\frac{1-k}{k}}\right)} \qquad (6)$$

η_e and η_c are the isentropic efficiencies of the expander and compressor, respectively, $T_{41} = T_4 / T_1$, and $P_{21} = P_2 / P_1$. Eq. (6) shows that the pressure ratio should no longer be as small as possible but that there is an optimum value, which is a function of η_e and η_c. Various techniques can be, and have been, employed to improve the air cycle's energy efficiency. For example: (1) The use of an additional internal heat exchanger has already been mentioned above. (2) Increasing the isentropic efficiencies of the turbomachinery, though this is difficult in practice since real devices already operate with high efficiency. (3) Multistage compression with intercooling, though at the expense of increased complexity, costs, and weight. (4) The use of moist air to improve the compression efficiency. However, these techniques do not address the performance of gas-to-gas heat exchangers. Their relatively poor convection heat transfer coefficients require larger surface areas than heat exchangers used in the vapor compression cycle. In general, however, the air cycle suffers from poor energy efficiency at higher refrigeration temperatures. Henatsch and Kauffled (1992) cite an example of transport freezing where the primary source COP of an actual vapor compression system at storage room temperatures of 0°C, –15°C, and –28.5°C is approximately 1.4, 1.06, and 0.7, respectively. For the same storage room temperatures, their calculated COPs for an air cycle are 0.9, 0.78, and 0.7, respectively. Implying that the air cycle may be competitive on an efficiency basis for very low evaporation temperatures. While it is true that the air cycle's energy efficiency is low, it might make sense in some special applications. For example, in aircraft cooling the benefit gained by not adding additional weight results in a lower required energy input (i.e., amount of fuel burned).

CONCLUSIONS

The current trend by Governments to restrict the use of HFCs in lieu of 'natural refrigerants' is premature and most likely counterproductive as far as the environment is concerned. This is particularly true when one considers that the global warming impact of refrigerants is estimated to be 2% of all manmade pollutants (Houghton et al., 1995). The global warming crisis differs from the ozone crisis in the heat pump system efficiency is the crucial factor for minimizing its life-long environmental impact. For over a decade, the industry has been using a quantitative measure that combines the direct and the indirect global warming impact. This measure, known as TEWI, has demonstrated that for virtually all systems the least amount of efficiency degradation will result in more atmosphere damage than the escaped refrigerant does, over the system lifetime. A performance standard based on TEWI, or some lifetime CO_2 equivalent measure, is the best approach to minimizing the environmental impact of future heat pumping systems. In order to achieve the best efficiency possible, heat pump system designers need maximum flexibility in their choice of materials and components. Eliminating refrigerants from their list of options can never lead to an optimal environmental solution.

Research into new technological concepts, of any sort, usually begins with computer simulations and laboratory prototypes. Progress of prototype development is often measured against the state-of-the-art production model it is intended to replace. This production model has, however, been designed with certain constraints that usually include safety, economic, durability, serviceability, etc. factors to which the prototype has not been subjected. In addition, laboratories, that are not associated with manufacturers, seldom limit their prototype development to cost effective components or system designs. The manufacturer designs the production model to meet all of the required standards for the minimum production cost. This is a necessity brought on by the competitive nature of this or of any other low technology industry within the free enterprise system. Of course, higher performance, more costly models are also produced; but history tells us that it is the lowest cost model that dominates the sales of virtually any heat pump system market. Thus if a prototype will inherently require a greater cost (e.g., an additional heat exchanger) to meet the performance of an existing production model, then the equivalent cost benefit must be given to any existing production system for the purposes of improving its performance. For example, the equivalent heat exchanger surface area could be added to the production model's evaporator and/or condenser, which would surely improve its current measured performance. Such improved performance estimates can be and have been done by reverse engineering techniques used by contractors for the U.S. Department of Energy (DOE/CS-0166, 1980).

Clearly it is technically possible to replace some of the current HFC systems with natural fluid systems. Ammonia may be able to expand to somewhat smaller systems. Hydrocarbons may be used in somewhat larger systems than domestic refrigerators. Carbon dioxide, in its transcritical cycle, may be made competitive in selected applications, where criteria other than performance are important. Even air has potential for transportation systems where system weight is a factor in the overall system efficiency. The real question is, however, will their respective production models be as environmentally friendly as their cost equivalent HFC alternatives? For example, the minimum USA standard for residential air conditioners is a seasonal COP of 2.9; however, some of the more expensive units of the same capacity have a COP of 4.1. If the global warming problem is so critical, then it is the upper limit value that should be the goal. When governments simply set a specification standard, such as 'thou shalt not use this or that HFC', there will be no incentive for the industry to design its lowest cost production systems to any performance level other than the current minimum. In fact competition will ensure such, as it does now. On the other hand, an environmental performance standard could be set to an impact level that would actually improve what we have today.

REFERENCES

Didion, D.A., 1999, The Application of HFCs as Refrigerants, *Proceedings of the Centenary Conference of the Institute of Refrigeration*, London, SW1H 9JJ.

Domanski, P., 1997, Theoretical Evaluation of the Vapor Compression Cycle With a Liquid-Line/Suction-Line Heat Exchanger, Economizer, and Ejector, *NISTIR 5606*, The National Institute of Standards and Technology, Gaithersburg, MD.

Engineering Analysis: DOE/CS-0166, 1980, U.S. Department of Energy, Assistant Secretary for Conservation and Solar Energy Office of Buildings and Community Systems, Washington, DC.

Fairchild, P., et.al., 1991, Total Equivalent Global Warming Impact: Combining Energy and Fluorocarbon Emissions Effects, *Proceedings of the International CFC and Halon Alternatives Conference*, Baltimore, MD.

Henatsch A. and Kauffled, M., 1992, Joule Cycle, in *Compression Cycles for Environmentally Acceptable Refrigeration, Air Conditioning and Heat Pump Systems*, International Institute of Refrigeration, Paris.

Houghton, L.G., et.al., 1995, Climate Change 1994: Radiative Forcing of Climate Change and an Evaluation of the IPCC IS92 Emissions Scenarios, Cambridge University Press.

McLinden, M. O., 1988, Thermodynamic Evaluation of Refrigerants in the Vapour Compression Cycle Using Reduced Properties, *Int. J. Refig.*, vol.11, no.3, pp.134-143.

Kates, R.W., et.al., National Academy Report, 1999, *Our Common Journey: a Transition Toward Sustainability*, National Academy Press, Washington, D.C., USA.

Pettersen, J., 1999, Carbon Dioxide (CO_2) as a Primary Refrigerant, *Proceedings of the Centenary Conference of the Institute of Refrigeration*, London, SW1H 9JJ.

Pillis, J.W., 1993, Expanding Ammonia Usage in Air Conditioning, *Proceedings of the ASHRAE/NIST Refrigerants Conference: R-22/R-502 Alternatives*, American Society of Heating, Refrigerating, and Air Conditioning Engineers, Atlanta, GA.

Starner, K.E., 1993, Heat Exchangers for Ammonia Water Chillers: Design Considerations and Research Needs, *Proceedings of the ASHRAE/NIST Refrigerants Conference: R-22/R-502 Alternatives*, American Society of Heating, Refrigerating, and Air Conditioning Engineers, Atlanta, GA.

Stera, A.C., 1993, Developments in Transportation of Chilled Produce by Sea and Air, *IIR Conference Proceedings: Cold Chain Refrigeration Equipment by Design*, International Institute of Refrigeration, Paris.

Stoecker, W., 1989, Expanded Applications for Ammonia – Coping with Releases to the Atmosphere, *CFCs: Today,s Options – Tomorrow's Solutions, Proceedings of ASHRAE's 1989 CFC Technology Conference*, ASHRAE, Atlanta, GA.

FIGURES

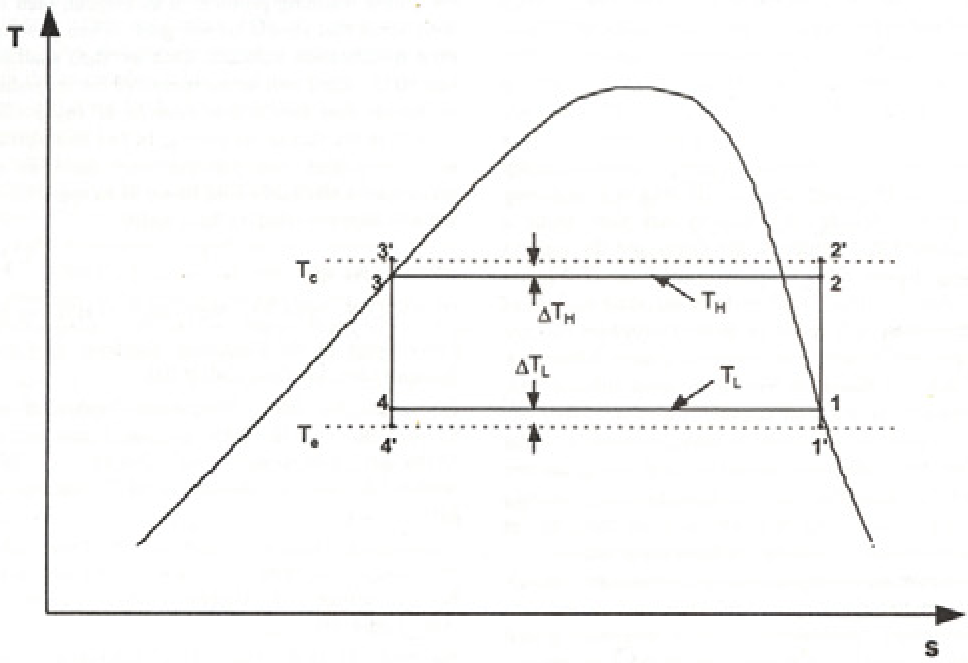

Figure 1a. Carnot Cycle (1-2-3-4) and a cycle with irreversible heat transfer processes (1'-2'-3'-4').

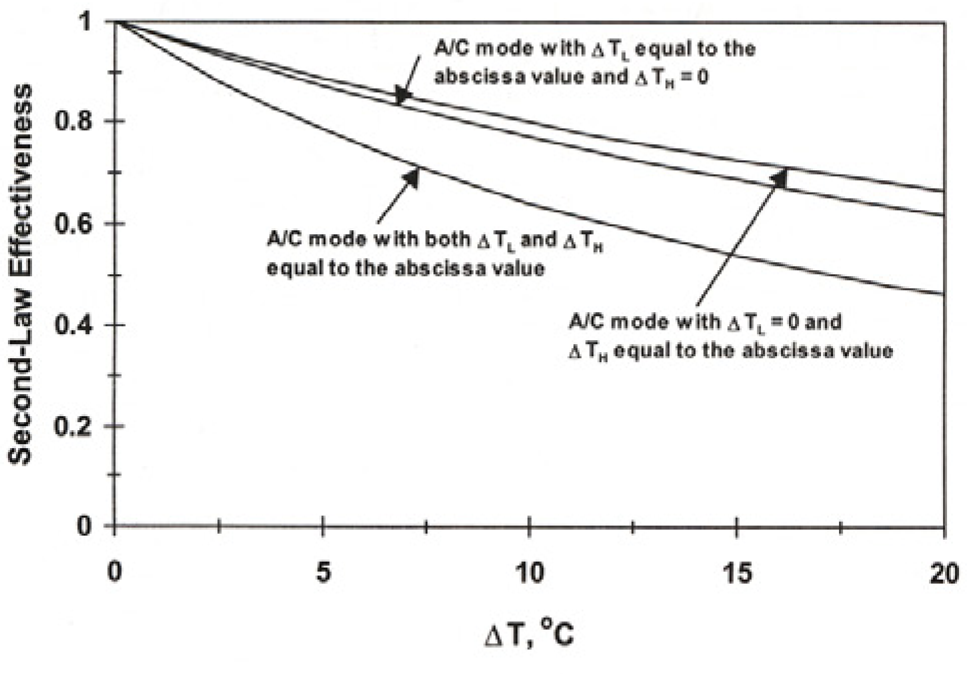

Figure 1b. Impact of increased temperature differences on COP.

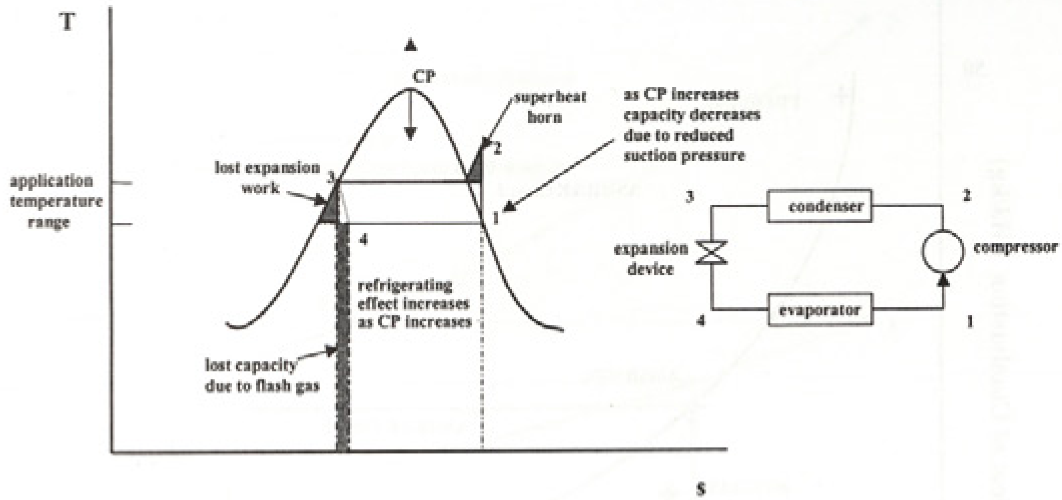

Figure 2. Vapor compression cycle.

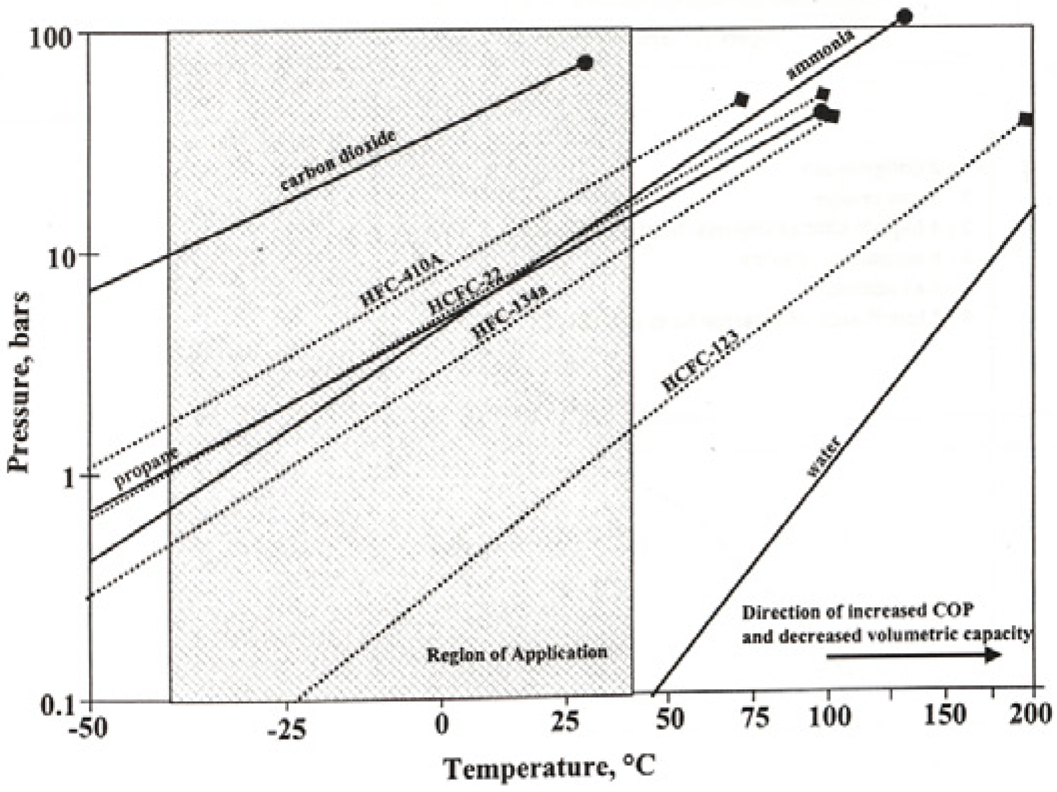

Figure 3. Vapor pressures of various refrigerants.

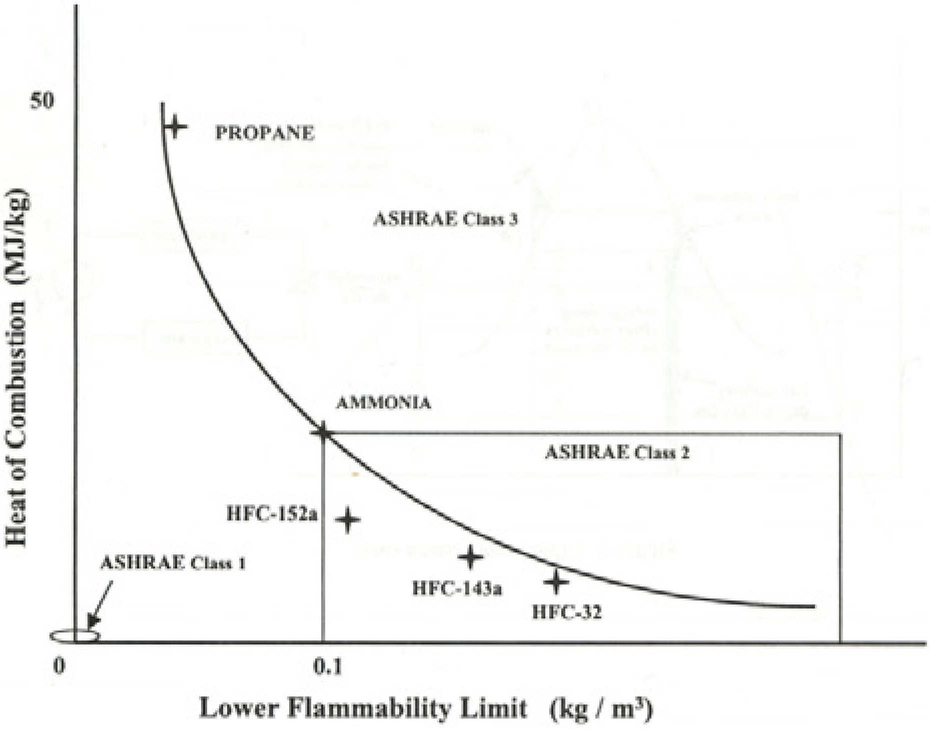

Figure 4. Flammability trends in refrigerants.

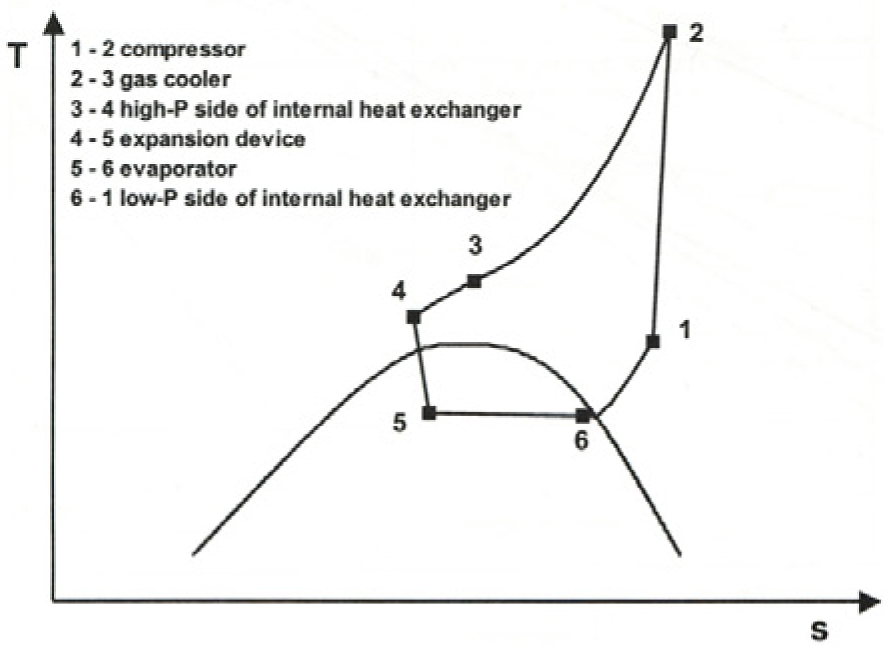

Figure 5. CO_2 transcritical cycle.

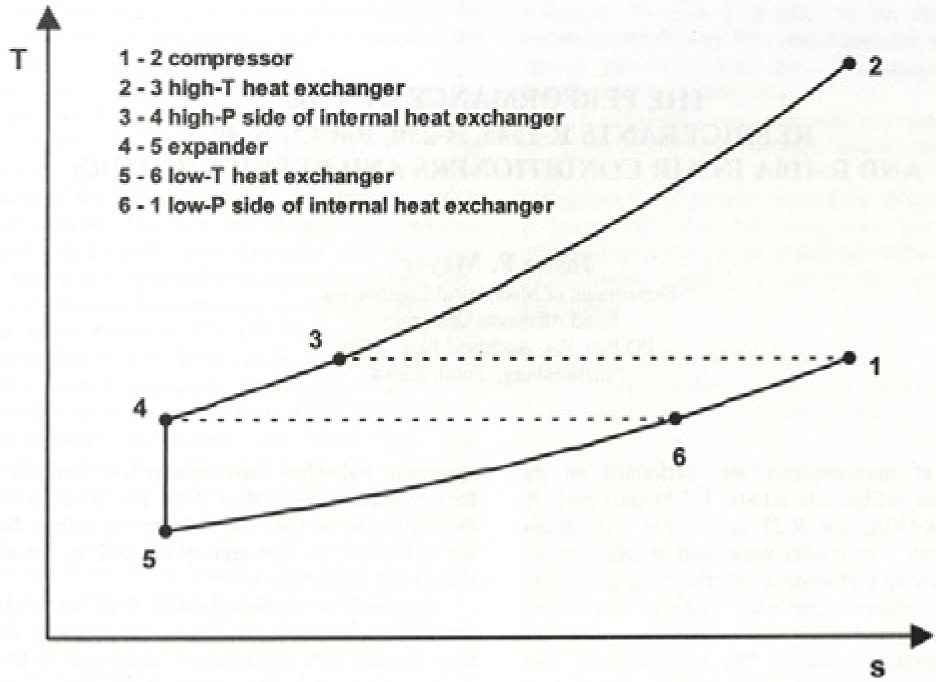

Figure 6. Air refrigeration cycle with an internal heat exchanger.