2006

Performance of High Pressure Air Cycle in Comparison With Synthetic and Natural Vapor Compression Refrigeration Cycles

Horst Kruse  
FKW-GmbH

Hans Rüssmann  
FKW-GmbH

Rainer Jakobs  
IZW

Follow this and additional works at: http://docs.lib.purdue.edu/iracc
Performance of High Pressure Air Cycle in Comparison with Synthetic and Natural Vapor Compression Refrigeration Cycles

Horst KRUSE¹, Hans RÜSSMANN², Rainer JAKOBS³

¹,² FKW – Forschungszentrum für Kältetechnik und Wärmepumpen GmbH
Weidendamm 12-14, D-30167 Hannover, Germany
Tel: +49–511–167475–0, E-mail: e-mail@fkw-hannover.de

³ IZW – Informationszentrum Wärmepumpen und Kältetechnik e.V.
Welfengarten 1a, D-30167 Hannover, Germany
Tel: +49-511-167475-15, E-mail: email@izw-online.de

ABSTRACT
Concerning the use of HFCs as substitutes for CFCs and HCFCs a major concern in applications with remarkable leakages of refrigerant is its contribution to global warming. Therefore, the attention has been drawn to the use of natural working fluids, like carbon dioxide, hydrocarbons, water and air.

Air had been already used in the range of air conditioning technology. Air as refrigerants is standard in air conditioning of airplanes and also in the Inter-City-Express 3 of the German Railways. A research project of air cycle technology for transport refrigeration had been already reported by FKW at the Purdue Conference of 1996.

In order to apply high pressure equipment as developed recently for transcritical carbon dioxide systems the single-stage high pressure air cycle process has been investigated theoretically as an alternative solution to conventional vapor compression systems with HFC-refrigerants and even to CO2-systems. An evaluation of the high pressure air cycle system has been done concerning their energetic and volumetric performances.

1. INTRODUCTION
During the Purdue and IIR Conferences of the last 10 years some technical presentations dealt with the air cycle in order to evaluate the application of the natural refrigerant air under the new environmental challenges of ozone depletion and global warming. It is well known that the energetic efficiency of the theoretical Joule-Cycle with two isentrops and two isobars shows a lower COP than the theoretical vapor compressor cycle based on the Carnot Cycle with two isentrops and two isotherms as shown in Figure 1.

On the other hand it is also well known that most of the cooling and heat rejection tasks for refrigeration machines are not isothermal, because heat source and heat sink are not of infinity but are most often mass flows of secondary fluids like air or water or brine. Therefore, also the adaptation of the refrigeration process to the task plays an important role for the COP of the system whereby non-isothermal refrigeration processes could be advantageous.

Further on, nowadays not only the energy consumption of a refrigerating or air conditioning system is of importance but also the total greenhouse gas emissions caused by the refrigerant contribution by its Global Warming Potential and by the energy consumption of the system leading to carbon dioxide emission in fossil fuel power conversion plants. If a refrigerant with GWP = 0 like air can be used in certain RAC applications the Total Equivalent Warming
Impact could be lower at high refrigerant leakage, possible like in transport refrigeration and air conditioning. Therefore, besides of the long time application of the air cycle in aircraft air conditioning systems the German Railway in the 90’s of last century applied air cycle systems in its high speed trains using aircraft expansion machines in closed instead of open systems there. Therefore, further air cycle R & D work at FKW dealt mainly with transport refrigeration and air conditioning but also for building air conditioning and food freezing purposes /4/, /7/. The publication about air cycle systems in the last 10 years, Purdue and IIR Conferences are shown in Table 1.

Table 1: Overview of citations for air cycle systems

<table>
<thead>
<tr>
<th>Name</th>
<th>Year</th>
<th>Application</th>
<th>Theor.</th>
<th>Exp.</th>
<th>Source</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engelking, Kruse</td>
<td>1996</td>
<td>Development of air cycle technology for transport refrigeration</td>
<td>x</td>
<td>x</td>
<td>/4/</td>
</tr>
<tr>
<td>Halm et al.</td>
<td>2000</td>
<td>Air cycle technology used for truck air conditioning</td>
<td>x</td>
<td>x</td>
<td>/7/</td>
</tr>
<tr>
<td>Andou and Okuda</td>
<td>2002</td>
<td>Development of air refrigerant system</td>
<td>x</td>
<td>x</td>
<td>/1/</td>
</tr>
<tr>
<td>Hwang et al.</td>
<td>2003</td>
<td>Performance Analysis of advanced air cycle Refrigeration system</td>
<td>x</td>
<td>x</td>
<td>/8/</td>
</tr>
<tr>
<td>Gigiel et al.</td>
<td>2004</td>
<td>Rapid freezing of food in an open air cycle freezer</td>
<td>x</td>
<td>x</td>
<td>/6/</td>
</tr>
<tr>
<td>Doran et al.</td>
<td>2004</td>
<td>Evaluating the air cycle as a refrigerant free alternative for temperature controlled road transport</td>
<td>x</td>
<td></td>
<td>/3/</td>
</tr>
</tbody>
</table>

The main goal of the air cycle research under the aspect of TEWI was the improvement of the energetic efficiency, for which the compressor and expander efficiencies are very important, especially the latter one. Therefore, FKW’s work was strongly directed to the application of the so called Pressure Waves Machine. All the R & D work done up to now was mainly directed towards the one or both sides open cycle in order to avoid as much as possible heat exchanger losses.

On the other hand, the low fluid density of the air at low pressures and hence the low density of capacity and power of the systems leads to a relatively high influence of parasitic cycle losses like friction etc., so that the question should be raised, if a closed system with much higher pressures than 2,5 bar as those at the German Railways systems should be used, according to a very early proposal of Carl von Linde for a closed Joule-Thompson Process. The application of carbon dioxide nowadays in transcritical cycles has led to the development of components like compressors, heat exchangers and valves etc. for pressures up to 120 bars.

The question is, if with such components for air cycles an energy dense air refrigeration system could be advantageous and at least compete with CO2 or at temperatures below its triple point be a possible substitute for the hydrofluorcarbons like R23. To answer this question FKW conducted a study for the application of a high pressure closed air cycle to be used in cooling or freezing applications between -18°C and -110°C, from cold store and container temperatures down to temperatures of process cooling like freeze drying or material investigations between -60°C and -80°C and medical applications in cold chambers for rheumatic treatment in hospitals.

2. ENERGETIC AND VOLUMETRIC COMPARISON OF HIGH PRESSURE AIR CYCLE WITH HFC AND CO2 VAPOR REFRIGERATION UNITS IN DIFFERENT AIR CONDITIONING AND REFRIGERATION APPLICATIONS

The evaluation has been performed over the whole range of application temperatures for refrigeration systems between -18°C and -110°C. Also, the plant complexity on the basis of the number of the system parts has been evaluated for comparison.

For the different cooling ranges FKW models have been used for calculations with suitable assumptions for the expansion machines (see paper C138 of this conference).

A development of air expansion machines for high pressures is needed, whereby at least the application of a transcritical carbon dioxide expansion machine as developed by FKW could be taken into account.
2.1 Medium temperature applications from -18 to -30°C

2.1.1 E.g. container refrigeration, refrigeration of cold stores at -18°C

The following boundary conditions apply for the comparison at a temperature of e.g. -18°C:

Outside temperature: \( T_{amb} = +38°C \)
Inside temperature: \( T_{in} = -18°C \)

A conventional refrigerating plant was applied for the cold vapor process, consisting of 5 main components, with the usual temperatures and pressures for those plants at the several system components.

Single stage refrigerating plant R134a, R404A, CO₂ (5 main components):
- with 1 compressor,
- with 2 external heat exchangers,
- with 1 throttle valve,
- and 1 internal heat exchanger.

The single stage high pressure air expansion process between 50 bar und 200 bar was chosen as an alternative concept of calculation by FKW. The design is identical to the conventional refrigerating plant, but an expansion machine is used instead of the throttle, provided that there are expansion machines for high pressures available or will be developed e.g. like a pressure waves machine. The construction effort of those cold air refrigerating plants can be equal or a little less in comparison to a conventional refrigerating plant, if they work with high pressures, because at higher pressures the volume flow of the air being necessary for a special refrigeration performance decreases.

Furthermore, following assumptions apply for the comparison:

Conventional refrigerating plant:
- Isentropic efficiency of the compressor: \( \eta_{is,comp} = f(\text{pressure ratio}) \)
- Superheating at the compressor: \( \Delta T_{sh} = 6 K \)
- Subcooling at the condenser \( \Delta T_{sub} = 5 K \)

Single stage high pressure expansion process:
- Isentropic efficiency of the compressor: \( \eta_{is,comp} = 0.7 \) (pressure ratio \( \approx \) const)
- Isentropic efficiency of the expander: \( \eta_{is,ex} = 0.7..0.8 \)

Essential for both refrigerating plants:
- Temperature difference at condenser or gas cooler: \( \Delta T_{HX} = 5 K \)
- Temperature difference at the internal heat exchanger: \( \Delta T_{HX} = 5 K \)
- Temperature difference at the evaporator: \( \Delta T_{HX} = 10 K \)

In Figure 2 the respective schemes of the individual systems are shown, in which the expander is shown independently of its type. In the case of a pressure waves machine a part of the compression could be realized by its compression side in Bootstrap-mode of the system, the other one by an additional auxiliary compressor, e.g. as a scroll compressor.

![Figure 2: Single stage refrigerating plant and a closed single stage air refrigerating plant with internal heat exchanger and expander](image)
The best coefficients of performance result in this case from a conventional R134a-plant and from a conventional R404A-plant, followed by a single stage plant with the high pressure air refrigeration process with isentropic expander efficiency of $\eta_{Ex} = 0.8$ and $\eta_{Ex} = 0.7$ better than a conventional CO2-plant, see Figure 3, left.

In Figure 3, right, the comparison of the volume flows is shown, relating to the R134a-refrigerating plant. The volume flow of the air cycle hardly depends on the isentropic expander efficiency, so that the volume flow of the high pressure air refrigerating plant in the diagram is valid for the chosen range of the expander efficiencies. The volume flows in a high pressure air refrigeration cycle in case of frozen mode 17% of such of a R134a-cycle and nearly 1/3 of that of a R404A-cycle. A comparison of the air cycle with the CO2-cycle shows that the volume flows are about the same.

In case of the high pressure air refrigeration cycle the low pressure was chosen to 90 bar and the pressure ratio nearly to 1.5, by which the optimal values for the coefficient of performance can be achieved. A pressure ratio of 1.5 would be perfect in case of a pressure waves machine working like that of the shock wave. Thereby it may be possible that at such small pressure ratios the compressor efficiency could be better than 0.7 as assumed which would result in a slightly higher coefficient of performance. It is mentionable, that the compressor outlet temperatures in a high pressure air refrigeration cycle are close to 100°C.

![Figure 3: Comparison of COP of refrigeration and volume flow, relating to R134a at -18°C](image)

**2.1.2 E.g. Frozen food, container refrigeration at -30°C**

Frozen food in stores or supermarkets at temperatures of e.g. -30°C.

The following boundary conditions apply for the comparison:

- Outside temperature: $t_{amb} = +38°C$
- Inside Temperature: $t_{in} = -30°C$

Again a conventional refrigerating plant was applied for the cold vapor process as described in chapter 2.1.1.

The best coefficients of performance result in this case from a conventional R404A-plant and from a conventional R134a-plant, followed by a single stage plant with the high pressure air refrigeration process at first and followed by a conventional CO2-plant, see Figure 4, left.

It is significant that the single stage CO2-cycle for the temperature range of -18°C to -30°C is not competitive concerning its energy compared with the high pressure cold air cycle.

On the right side of the Figure 4 the comparison of the volume flow is shown, relating to the R134a-refrigerating plant. The volume flow of the air cycle hardly depends on the isentropic expander efficiency, so that the volume...
flows of the high pressure air refrigerating plant in the diagram are valid for the chosen range of the expander efficiencies.

The volume flows in a high pressure air refrigeration cycle are in case of a conventional cycle essential less, 9\% of a R134a-cycle and nearly 4/5 less than that of a R404A-cycle. A comparison of the air cycle with the CO\(_2\)-cycle shows that the volume flows of the high pressure air cycle is down to 3.5 times lower as that of the CO\(_2\)-cycle.

CO\(_2\) from -30°C to -50°C is energetic better suitable in CO\(_2\)/NH\(_3\)-cascade system, since it goes only through a subcritical process.

\begin{figure}
\centering
\includegraphics[width=\textwidth]{fig4}
\caption{Comparison of COP of refrigeration und volume flow, relating to R134a at -30°C}
\end{figure}

\subsection{2.2 Low temperature applications from -50°C to -120°C, e.g. bio-stocks, conservation of blood, tests of material, rheumatism chambers}

To analyze the qualification of the high pressure process by using air as refrigerant, in the following the range of lower temperatures will be compared with conventional refrigerating plants in cascade, in which case CO\(_2\) cannot be used because of the triple point of -56°C.

\subsubsection{2.2.1 E.g. bio-stocks, conservation of blood between -50°C and -60°C}

In this case a two-stage cascade consisting of 7 main components can be used, for less capacious plants usually with R23/R134a or for capacious plants up to -54°C, too, with CO\(_2\) in the lower and NH\(_3\) in the higher stage, see Figure 5.

\begin{figure}
\centering
\includegraphics[width=\textwidth]{fig5}
\caption{Two-stage cascade system and a closed single-stage air refrigeration plant with internal heat exchanger and expander}
\end{figure}
The result in comparison with cold air process with single stage compression and expansion, which is really a simpler and cheaper system, is shown in Figure 6 and points out that the CO₂/NH₃-cascade at compressor efficiencies of 0.7 reaches a coefficient of performance of 1.15. The cold air process between pressures of 90 and 150 bar in contrast reaches a coefficient of 0.45 or 0.52 at efficiencies of the compressor and expander of 0.7 bzw. 0.8. This is 46% of the conventional technology, and it works both with synthetic and with natural substances for the refrigerant.

The volume flow of the lower stage of the conventional vapor compression system R23/R134a-cascade is in comparison to a cold air system lower, in comparison to a system with CO₂ higher.

2.2.2 E.g. test of materials at -70°C

The technical comparison has the following boundary conditions:

Outside temperature: \( t_{\text{amb}} = +38°C \)

Inside temperature: \( t_{\text{In}} = -70°C \)

Again conventional cascade systems were applied for the cold vapor process as described in chapter 2.2.1. At this low temperature the application of CO₂ in the lower cascade stage is not possible, because the triple point of CO₂ is -56.6°C. Anymore, if HFC-refrigerants should not be used, in this case – according to studies of FKW Hannover /5/ and the ILK Dresden /9/ – CO₂ can be exchanged by N₂O in the lower stage, because its triple point is -90.8°C.

The design of cold air refrigeration plants at such lower temperatures doesn’t conform anymore to the conventional refrigeration plant. Here the construction effort in comparison to a conventional cascade system is much lower with a really simpler and cheaper systems engineering.

**Figure 6: Comparison of COP of refrigeration und volume flow, relating to R134a at -50°C**

2.2.2 E.g. test of materials at -70°C

The technical comparison has the following boundary conditions:

Outside temperature: \( t_{\text{amb}} = +38°C \)

Inside temperature: \( t_{\text{In}} = -70°C \)

Again conventional cascade systems were applied for the cold vapor process as described in chapter 2.2.1. At this low temperature the application of CO₂ in the lower cascade stage is not possible, because the triple point of CO₂ is -56.6°C. Anymore, if HFC-refrigerants should not be used, in this case – according to studies of FKW Hannover /5/ and the ILK Dresden /9/ – CO₂ can be exchanged by N₂O in the lower stage, because its triple point is -90.8°C.

The design of cold air refrigeration plants at such lower temperatures doesn’t conform anymore to the conventional refrigeration plant. Here the construction effort in comparison to a conventional cascade system is much lower with a really simpler and cheaper systems engineering.

**Figure 7: Two-stage cascade system and a closed single-stage air refrigeration plant with internal heat exchanger and expander**
The result in comparison with the cold air process with single stage compression and expansion is shown in Figure 8, left, and points up that the N₂O/NH₃-cascade similar to the R23/R134a-system reaches a coefficient of performance of 0.8. The cold air process between pressures of 90 and 150 bar in contrast reaches efficiencies of the compressor and expander of 0.7 or, 0.8 a COP of 0.38 or 0.43. This is 54% of the conventional technology. The Figure 1 and Figure 8 show that the cold air process ameliorates energetically relative at lower temperatures compared to the cold vapor process, of which the efficiency highly decreases, but it doesn’t reach the energetic quality of conventional cascades with synthetic or natural refrigerants.

The volume flow of the lower stage of the conventional R23/R134a-cascade vapor compression system in comparison to a cold air system is more than twice as high and, too, in comparison to a system with N₂O it is higher. A comparison of the air system with the CO₂-system has a twice lower volume flow.

2.2.3 E.g. rheumatism chambers at -120°C
At about 150 K adequate -124°C the efficiencies of the cold vapor and cold air process are nearly equal (see Figure 1) For this reason both methods will be compared again: a real built system of conventional technology as three-stage cascade system – being often built for rheumatism chambers – in contrast to a single stage cold air system. The technical comparison has the following boundary conditions:
Outside temperature: \( t_{amb} = +25°C \)
Inside temperature: \( t_{In} = -110°C \)
A cascade system was applied for the cold vapor process, having 10 main components, with the usual temperatures and pressures at the several system components:
Three-stage cascade refrigeration plant R404A / R23 / R14 (10 main components):
- with 3 compressors,
- with 2 outside air heat exchangers,
- with 3 throttle valves,
- and 2 intercoolers.

As an alternative conception for the cold air process the closed single stage high pressure expansion process (5 main components) was analyzed:
- with 1 compressor,
- with 2 outside air heat exchanger,
- with 1 expansion machine,
- and 1 internal heat exchanger.

In this case the three-stage cascade uses the refrigerant R14, R23 and R404A. The cold air process is realized between the pressures of 50 bar and 125 bar and it utilizes only a single stage compressor with a pressure ratio of 2.5, whereby the discharge temperature is 155°C.
The three-stage cascade refrigerating plant – at an outside air temperature of +25°C and an inside temperature of -110°C – reaches a refrigeration coefficient of performance of 0.206 (100%). A cold air refrigeration plant reaches at efficiencies of the compressor and the expander of 0.7 and 0.8 respectively a coefficient of performance of 0.223 (120% in comparison with the cascade system) or 0.247 (108% in comparison with the cascade system), see Figure 10, left.

The R14-compressor delivers at 4 kW refrigeration capacity a volume flow of 40.5 m³/h as single stage reciprocating compressor with a maximal pressure of 25 bar. Furthermore the volume flows of the further two compressor stages have to be considered. In this case the volume flow of the cold air compressor in contrast to 40.5 m³/h (100%) at the R14-compressor is with 9.3 m³/h (only 23% in comparison to the cascade system) much lower, see Figure 10, right.

International Refrigeration and Air Conditioning Conference at Purdue, July 17-20, 2006
3. EXPANSION MACHINES

For the high pressure expansion process with cold air the efficiency of the compressor and of the expansion machine are very significant, in particular that of the expansion machine.

3.1 Pressure Waves Machine

Turbo expansion machines are used at present mainly for middle to large flow rates. With decreasing flow rates with higher pressures the isentropic efficiency for turbo compression and turbo expansion machines drop strongly, as is represented in Figure 11. Small turbo machines would be not suitable because of the small volume flow and hence efficiency for high pressure air processes.

As alternative to turbo machines for small flow rates FKW intensively examined the pressure waves machines which was reported in detail at the Purdue Conference in 1996 and 2000 (Figure 13).

In other gas compression and expansion tasks pressure waves machines are already developed for pressures up to 70 bar, but for closed high pressure air systems a development for pressures up to 150 bar is needed and possible.

![Figure 11: Isentropic efficiencies for turbo compressors (left) and turbo expanders (right)](image1)

![Figure 12: Pressure Waves Machine applicable for cool air processes](image2)

3.2 Reciprocating compressor-expander-unit

Another alternative is a reciprocating compressor-expander-unit which was already investigated and developed at FKW for transcritical CO₂ refrigeration systems, see Figure 13, right. The reciprocating CO₂ expansion cylinder works without valve or electrical or mechanical slider control as designed by Doll and Eder /2/ for cryogenic helium expansion. Closed high pressure air cycles can work also with oil lubricated machines and therefore in contrary to Doll and Eder can have piston rings for better cylinder sealing.

This unit achieved in the superheated single-phase gas region of CO₂-cycles expansion efficiencies of nearly 0,5. It would be interesting to employ and develop further such a simple expansion machine towards higher efficiencies for an effective high pressure air cycle. Thereby, a high pressure expansion process for a cold air refrigeration system would be possible producing the required useful temperatures with high efficiency and low construction effort.

![Figure 13: Reciprocating compressor-expander-unit applicable for cool air processes](image3)
4. CONCLUSIONS

It has been shown in the beginning that the air cycle is more favorable at lower than higher cooling temperatures. For applications with temperatures lower than 100°C cold gas expansion machines have an energetically more favorable behavior in relation to cold vapor machines, as shown in Figure 1. Therefore, for very low temperature applications from -50°C to 100°C where two or three stage cascades for conventional systems are applied for the cold vapor process the closed single stage high pressure air expansion process because of its fewer main components and lower flow rates is an alternative for the substitution of HFCs. For medium temperature application like container refrigeration (-18°C to -30°C) the closed single stage high pressure air expansion process could be an alternative for the substitution of HFCs having higher efficiencies and lower volume flow rates than the CO₂-one-stage cycle. Further on, air everywhere is available which can be of importance for transport applications with high leakage rates.

For transport air conditioning the closed high air pressure system is not particularly suitable as experienced at such a system for the ICE 3 of the German Railways because of higher energy consumption. For stationary air conditioning both sides open air cycle systems have the advantage that due to no external heat exchangers heat transfer losses can be minimized. It also allows an installation with fewer components than in conventional plants. At TNO in the Netherlands studies were conducted for the application of such an open air cycle system for the air conditioning of buildings /11/, which showed at 30°C 33% lower energy consumption than conventional vapor compressor water chillers with R134a.

The single-stage high pressure air cycle process theoretically is an alternative solution for CO₂-systems if conventional vapor compression systems with HFCs-refrigerants should be substituted, but an expansion machine for high pressures has to be developed. The development of such expanders of the pressure waves machine or reciprocating piston type is possible, as shown by FKW’s investigations.

REFERENCES