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International Refrigeration and Air Conditioning Conference

School of Mechanical Engineering

1994

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AN AIR CYCLE DESIGN CONCEPT FOR DOMESTIC OR SMALL COMMERCIAL REFRIGERATORS

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ABSTRACT

As CFC refrigerants are now being phased out as quickly as possible and as the full effects of replacement refrigerants on the environment may still be unknown, it would be ideal to find a refrigerant that was environmentally benign beyond question. Air is such a refrigerant. The characteristics of a small commercial refrigerator that uses R134a refrigerant are used as a basis for a preliminary design of an air cycle refrigerator that would have similar characteristics.

NOMENCLATURE

c.o.p.	coefficient of performance
c_p	specific heat at constant pressure
F_{isc}	isentropic enthalpy factor
F_{isc}^{w}	isentropic work ratio
h	specific enthalpy
h'	specific enthalpy after an ideal isentropic process
k	specific heat ratio, c_p/c_v
m	mass flow rate
p	pressure
q_L	specific cooling effect
Q_L	cooling effect
T'	temperature after an ideal isentropic process
T	temperature
W_{net}	net work input = compressor work - expander work
η_{isc}	isentropic efficiency of compressor
η_{ise}	isentropic efficiency of expander

INTRODUCTION

The damage caused to the ozone layer by CFC refrigerants has caused alarm in the world. For some years now a search for replacement refrigerants has been underway. The recently introduced substitute for R12, R134a, is believed not to damage the ozone layer, although it does still contribute to global warming. However, although this refrigerant appears to have no other negative effects on the environment, this has not yet been proven. Hence, it still makes sense to research the possible development of domestic or small commercial refrigerators that would use a refrigerant that most definitely has no negative effect on the environment. Air is this refrigerant.

There is no fundamental reason, according to the laws of thermodynamics, why a single phase refrigerant (such as air) should not function as well as a two phase refrigerant (such as R134a). This paper proposes a preliminary design for an air cycle refrigerator.

WHY AIR?

In the past, air was used extensively as a refrigerant; for example, in cold store applications and in ship-based systems. Air cycle refrigeration was superseded by vapour compression refrigeration, which was more efficient. However, as a result of developments in materials and methods, air cycle systems could possess the following advantages when compared with conventional vapour compression systems:

1. The working fluid (air) is non-toxic and environmentally benign beyond all question.
2. Air is free and available in unlimited quantities, unlike other refrigerants.
3. Air cycle equipment (e.g., as used in aircraft) can be very reliable and maintenance costs during its life span can be low.
4. As the pressures involved are low and as absolute leak-tightness is not required, air cycle refrigerators, in a developed form, could prove less expensive than vapour compression units.

The following are some disadvantages of air cycles:

1. Unless major advances are made in the design of air cycle components, extra energy may be consumed due to the lower c.o.p. of these systems.
2. Air cycle performance is highly sensitive to the efficiency of the compression and expansion processes. Vapour compression cycles are less sensitive to the isentropic efficiency of the compressor. This is because much of the heat rejection from the vapour cycle is associated with the latent heat of condensation of the refrigerant. Therefore the mean temperature of heat rejection is more closely associated with the discharge pressure than with the discharge temperature of the compressor. Compressor efficiency is often sacrificed in vapour compression refrigerators in favour of reduced size, weight, and cost [1].
3. Whereas in vapour compression cycles the potential expansion work is usually too small to justify its exploitation, in air cycles the expansion work is equivalent to a large part of the compression work and must be effectively recycled.

Given the advantages and disadvantages of air cycle and vapour compression systems, and especially in view of the fact that the environmental implications of any system are becoming more important, the development of a viable air cycle refrigerator merits consideration.

DESIGN SPECIFICATIONS

A small commercial refrigerator, which used refrigerant 134a, was investigated to determine its characteristics. It had a cabinet volume of 125 litres. Air was circulated around the cabinet by a full width, cross flow fan. The objective was to establish the required characteristics of an air cycle refrigerator to be used for domestic refrigeration or for the same commercial niche. A similar cooling effect, similar temperatures, a similar air circulation rate, and similar power consumption were required.

Following the analysis of the R134a system, it was decided to specify an air cycle with a temperature of 2°C (275.15 K) at entry to the cabinet and 8°C (281.15 K) at exit. There was to be pressure equalisation between the cabinet and the atmosphere. The mass flow rate through the system was to be 0.029 kg/s. The design atmospheric temperature was taken to be 20°C (293.15 K). A c.o.p. of about 1.9 was required to match the performance of the vapour compression system.

THERMODYNAMICS OF AIR CYCLE REFRIGERATION

The use of air as a refrigerant is based on the principle that when a gas expands adiabatically while doing work its final temperature at the new pressure is lower than the initial value. Air cycle refrigeration can be implemented in an open or closed cycle configuration. In an open cycle the "refrigerant," air, is used to cool the cabinet of the refrigerator directly without a heat exchanger.

The conventional air cycle is the reversed Brayton cycle, the ideal form of which is shown in Fig. 1. A low pressure ratio is required, as can be seen from the expression for the coefficient of performance of the ideal reversed Brayton cycle:

$$F_{\mathrm{isc}} = \frac{h_2' - h_1}{h_2 - h_1}. \text{ With a constant } c_p \text{ this reduces to } F_{\mathrm{isc}} = \frac{T_2' - T_1}{T_2 - T_1}. \tag{7}$$

$$F_{\mathrm{isc}}^{w} = \frac{\text{ideal work for isentropic compression}}{\text{actual work for non-adiabatic compression}}. \tag{8}$$

PROPOSED CYCLE DESIGN

To specify a suitable cycle for the air cycle refrigerator, each of the factors that could affect its performance was examined to consider if it was worthwhile to design specifically for that factor in the cycle. The temperatures into and out of the refrigerator cabinet were fixed due to the constraint of the air cycle refrigerator having the same cooling effect as the existing commercial refrigerator.

Pinch Point

It was necessary to quantify the pinch point temperature difference of the heat exchanger(s) to be used; i.e., the minimum temperature difference between the fluids across the heat-exchange surface. The greater the design pinch point temperature difference, the lower would be the c.o.p. It was decided that the design pinch point temperature difference would need to be 8 K.

Multi-stage Compression

It is difficult in practice to achieve a heat transfer rate during compression that is high enough to effect a significant reduction in work. A practicable alternative is to separate the work and heat transfer processes. Hence, the use of intercooling between stages of compression could be advantageous.

The improvement in the coefficient of performance was evaluated for multiple stage compression with intercooling. When a value of 0.8 for the isentropic efficiency of the compressors and of the expander was used it was found that a 4% rise in the c.o.p. was obtained by increasing the number of stages of compression from one to two. Increasing the number to three stages resulted in a further increase of 0.6%. With a value of 0.97 for the isentropic efficiency of the compressors and of the expander, an increase of 3.76% in the c.o.p. was obtained by increasing the number of compression stages from one to two. A further increase of 0.69% was obtained by increasing the number of compression stages to three.

These increases in c.o.p. would not justify the increased complexity of the plant. Hence, the use of single stage compression is proposed.

Addition of a Recuperator

The addition of a recuperator, which would transfer heat between the streams of air entering the expander and leaving the compressor, would lower the temperature before expansion and raise the temperature before compression. It would also have the effect of lowering the pressure ratio of the cycle (p_2 / p_1 or p_3 / p_4). However, this option proved ineffective as it caused a decrease in c.o.p. of 4.5%.

Intake of Ambient Air

An investigation was made to determine the effect of taking in ambient air to the compressor while discharging the exhaust air from the refrigerator at state 1 in Fig. 1. This would give a higher temperature before compression. The result was a decrease in c.o.p. of 3.5% because the increase in temperature before compression caused the temperature after compression and the work of compression to be increased as well. Hence this was not an advantageous cycle modification.

$$\text{c.o.p.} = \frac{\text{cooling effect}}{\text{net work input}} = \frac{Q_L}{W_{net}} = \frac{1}{\left(\frac{p_2}{p_1}\right)^{\frac{k}{(k-1)}} - 1}. \tag{1}$$

The cooling effect of the reversed Brayton cycle can be calculated as follows:

$$Q_L = q_L m, \text{ where } q_L = c_p(T_1 - T_4). \tag{2}$$

Although the reversed Brayton cycle is often regarded as the ideal for air cycle refrigeration, it is not the optimum thermodynamic cycle. A thermodynamically ideal air cycle for the application that has been defined would be the cycle shown in Fig. 3. This consists of two compression processes (one isentropic; and one isothermal, with heat rejection at the temperature of the surroundings), one isentropic expansion process, and heat acceptance at constant pressure. Actual cycles should be designed to approach this cycle, rather than the reversed Brayton cycle.

In a real cycle the compressor and expander are often regarded as adiabatic. The compression and expansion processes are then characterised by the isentropic efficiencies of these units. Therefore, the real cycle is often modelled as an ideal Brayton cycle modified by these efficiencies, as shown on the *T-s* diagram in Fig. 2.

The isentropic efficiency of the compressor is defined as follows:

$$\eta_{isc} = \frac{h_2' - h_1}{h_2 - h_1}. \text{ With a constant } c_p \text{ this reduces to } \eta_{isc} = \frac{T_2' - T_1}{T_2 - T_1}. \tag{3}$$

The isentropic efficiency of the expander is defined as:

$$\eta_{ise} = \frac{h_3 - h_4}{h_3 - h_4'}. \text{ With a constant } c_p \text{ this reduces to } \eta_{ise} = \frac{T_3 - T_4}{T_3 - T_4'}. \tag{4}$$

Also, for an adiabatic compressor the actual work input per unit mass is equivalent to the increase in the specific enthalpy. For an adiabatic expander the actual work output per unit mass is equivalent to the decrease in the specific enthalpy. Therefore

$$\eta_{isc} = \frac{\text{ideal work for isentropic compression}}{\text{actual work for adiabatic compression}} \tag{5}$$

and

$$\eta_{ise} = \frac{\text{actual work for adiabatic expansion}}{\text{ideal work for isentropic expansion}}. \tag{6}$$

It can be noted that an isentropic efficiency characterises (1) the exit state of the device and (2) the relationship between the ideal work for isentropic compression and the actual work.

Non-Adiabatic Compression

Heat rejection from the gas during compression is advantageous as it reduces the required work input. If there is heat rejection, the isentropic efficiency, which is defined in Equation 3, is not strictly appropriate. However, the right hand side of the same expression could be given a different name: the term *isentropic enthalpy factor*, F_{isc}, will be used to characterise the exit state of the compressor. The term *isentropic work ratio*, F_{isc}^{w}, will be used to characterise the relationship between the actual work input of a non-adiabatic compressor and the ideal isentropic work. In theory, both of these factors can have values greater than unity.

Effect of Pressure Drop Across the Heat Exchanger

There is a significant decrease in the c.o.p. as the pressure drop across the heat exchanger increases, as shown in Fig. 4. To achieve similar performance to a vapour compression refrigerator it was deemed necessary to specify a heat exchanger with a design pressure drop of 0.001 MPa.

Effect of Isentropic Efficiency

As can be seen from Fig. 5, the sensitivity of the c.o.p. to the isentropic efficiencies is particularly high when they are in the region of 0.9 to 1. It would be necessary to design a compressor and an expander with isentropic efficiencies of 0.97 in order to achieve similar performance levels to those of vapour compression refrigeration. For a non-adiabatic compressor, the isentropic work ratio, F^{w}_{isc}, would have to be 0.97—this goal can be more readily achieved than an isentropic efficiency of the same value.

The achievement of such high efficiencies would require a special compressor/expander unit. A single-shaft positive displacement machine is envisaged. Both the compressor and the expander would probably be of the rolling piston type. The efficiency specifications would probably require much bigger swept volumes, much larger valve ports, and lower speeds than in conventional rolling piston machines. The expander would be designed as an adiabatic device, whereas the compressor would be designed for maximum heat rejection during compression. Although the compressor/expander unit would be likely to be relatively large, the low pressure ratio of the air cycle would allow a much lighter form of construction than is presently used for vapour compression compressors.

The specifications given so far fully define the proposed air cycle. It is shown in Fig. 6 and has a c.o.p. of 2.02. By allowing for a motor efficiency of 89% this figure can be adjusted to give a c.o.p. value of 1.8 for the electrically driven air cycle refrigerator.

CONCLUSIONS

To achieve a viable c.o.p. for an air cycle refrigerator it will be necessary to design and develop a special compressor/expander unit with a high efficiency. If this unit and the heat exchanger for heat rejection can be realised according to the specifications that have been proposed then air cycle refrigerators could provide an environmentally benign alternative to conventional vapour compression refrigerators.

ACKNOWLEDGEMENT

The research described in this paper was funded in part by the Commission of the European Communities, within the JOULE II programme.

REFERENCE

[1] Gigiel A. J. - "Air as a Refrigerant," ACR Today, February, pp. 30-33, (1990).

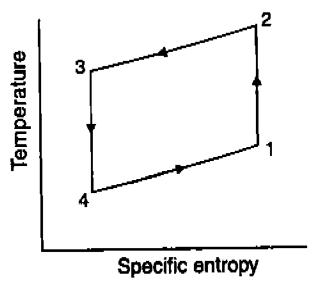

Fig. 1 : Ideal reversed Brayton cycle.

1-2: isentropic compression of air

2-3: heat rejection at constant pressure

3-4: isentropic expansion

4-1: heat acceptance at constant pressure.

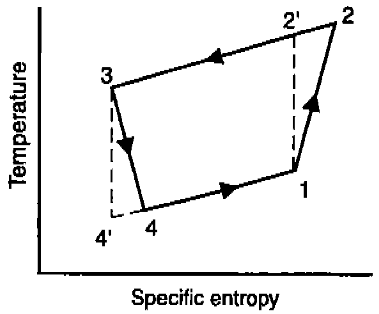

Fig. 2 : Real reversed Brayton cycle

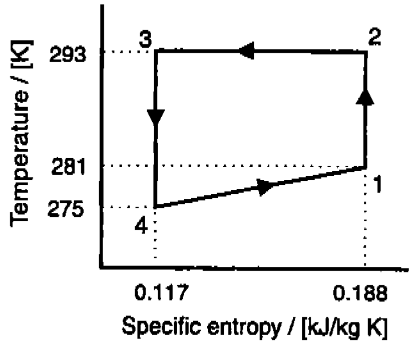

Fig. 3 : Optimum thermodynamic cycle

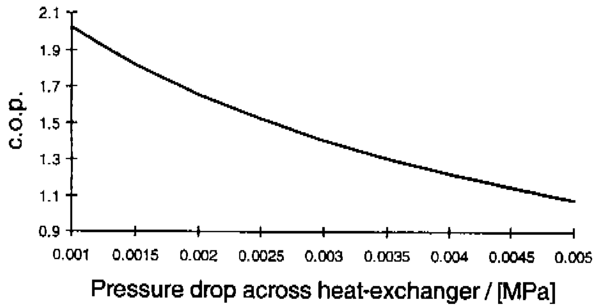

Fig. 4 : Pressure drop across the heat exchanger versus c.o.p. Calculations relating to this graph are based on an isentropic efficiency (of both compressor and expander) of 0.97. Points 1, 4, and 3 are fixed as show in Fig. 6.

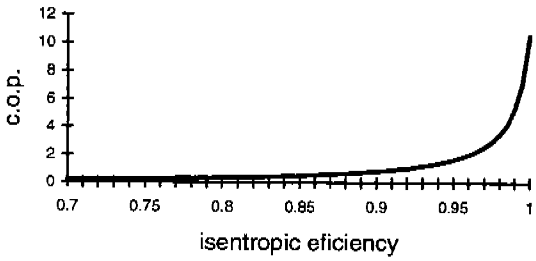

Fig. 5 : Effect of isentropic efficiency on the c.o.p. of the cycle. In the calculations underlying this graph it is assumed that the isentropic efficiencies of the compressor and expander are both the same. It is also assumed that the pressure drop across the heat exchanger is negligible. Points 1, 4, and 3 on the cycle are fixed at the conditions shown in Fig. 6.

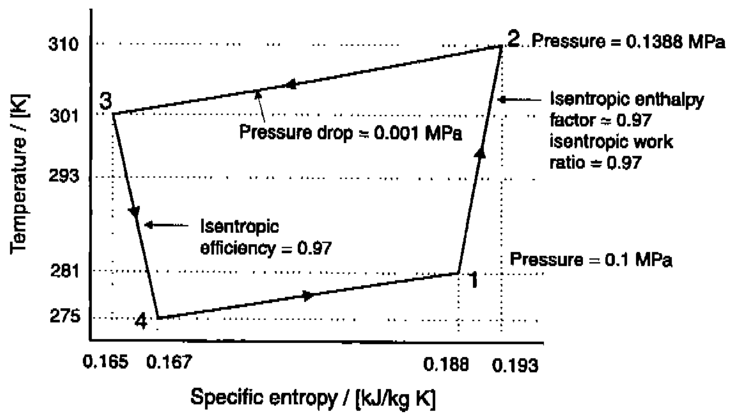

Fig. 6 : Proposed cycle