Potential of Carbon Dioxide in Two Refrigeration Applications Commonly Encountered in the Netherlands

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POTENTIAL OF CARBON DIOXIDE IN TWO REFRIGERATION APPLICATIONS COMMONLY ENCOUNTERED IN THE NETHERLANDS

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A B S T R A C T

An inventory study of the energy use in cold storage installations in The Netherlands showed that frozen storage and combined chilled storage and distribution are responsible for a substantial part of the energy consumption of the sector. Two reference complexes including a standard number of refrigerated rooms can represent these two categories of plants. These reference complexes are used to evaluate the potential of carbon dioxide transcritical or secondary cycles in these refrigeration applications.

First a second law analysis of the existing reference plants is presented identifying the contribution of the different components to the total irreversibility of the system. Then the comparable irreversibility losses are predicted when CO_2 is to be used in the same plants as a primary (transcritical) or secondary refrigerant.

Based on these results, the advantages and drawbacks of the two carbon dioxide options (transcritical / secondary) in such plants are compared taking the existing systems into account.

INTRODUCTION

An inventory study of the energy use in cold storage installations in The Netherlands (Bosma [1995]) showed that frozen storage, is responsible for about 60% of the energy consumption of the companies participating in the Dutch Cold Storage Association. Combined chilled storage and distribution represents only 10% of the energy consumption of the sector.

In most of these cold stores, HCFC 22 is used in the refrigeration plant. When looking for substitutes for this non-zero ODP medium, carbon dioxide appears as a candidate. Two options are then available: the application of a transcritical carbon dioxide cycle or the application of carbon dioxide in a secondary loop with, for instance, ammonia in the primary loop.

The purpose of this paper is to investigate the potential of carbon dioxide in reference cycles for the two types of cold storage applications representative for the sector in The Netherlands. The system performance calculation will be based on an exergy analysis. Exergy is a measure of the departure of the state of a fluid from that of the environment and is defined as (Moran & Shapiro [1993]):

$$ex = (h - h_o) - T_o (s - s_o) + c^2/2 + gz \tag{1}$$

The terms h_o and s_o represent the enthalpy and entropy values at the environmental temperature T_o and pressure p_o. Mostly the effects of motion and gravity can be neglected and eq. (1) reduces to

$$ex = (h - h_o) - T_o (s - s_o) \tag{2}$$

The exergy destruction within a control volume can be predicted from

$$\Delta Ex = \Sigma\, \dot{m}_i ex_i - \Sigma\, \dot{m}_e ex_e - \dot{Q}\,(1-T_o/T) - \dot{W} \tag{3}$$

The first two terms on the right-hand side account for the sums of the time rates of exergy transfer accompanying mass flow at the inlet and outlet, respectively. The third term represents the time rate of exergy transfer accompanying heat transfer at a constant temperature T. The last term represents the time rate of mechanical work transfer.

The analysis is based on the application of eq. (3) for control volumes enclosing each of the components of the system. Kinetic and potential energy changes are neglected.

FROZEN STORAGE

Reference application

The reference application is a frozen storage complex with a relative large number of transport movements per day: the door of the frozen storage is open during about 1.2 hours per day (10% of the working time), resulting in a relative large latent load in the air coolers. The complex consists of a large storage (5600 m^3) where 5 air coolers with 100 m^2 and 2 fans each have been mounted. The refrigeration cycle consists of an 8 cylinder two-stage reciprocating compressor (6 cylinders in the low stage side and 2 cylinders in the high stage side) with a cooling capacity of 70 kW (at -32°C evaporating temperature and 30°C condensing temperature). The refrigerant is HCFC 22. The condenser is air cooled with an external surface of 885 m^2. A condensation pressure controler is applied keeping the condensation temperature at 30°C. The cycle is a two-stage cycle with interstage cooling with an economizer.
For the conditions studied, the product enters the frozen storage with a temperature of -23°C. In total there are 50 ton of frozen product in the store. The installation is operating with on/off cycles since it has been designed to eventually cool down the product from -17°C to -23°C.

Exergy analysis HCFC 22 system

A simulation model for the system described above has been implemented in Simulink. With this model the exergy values at the different points of the refrigerating cycle and of the air in the freezing store have been calculated. The exergy losses for the different components of the system have then been calculated as proposed, for instance, by Auracher [1981], according to eq. (3). As these values are time dependent, the average values for 24 hours of operation have been calculated.

The dead state temperature T_0 has been taken as the thermodynamic averaged heat sink temperature and the dead state pressure p_0 has been taken as the ambient pressure. The exergy input to the system connected with the airflow through the condenser is negligible. The exergy input connected with the airflow through the air coolers is relatively large but not shown in fig. 1. Mainly three other exergy flows enter the system: the exergy flows connected with the electric energy supply to the condenser (3%) and evaporator (1%) fans and to the compressor drive (96%). The cooling effect corresponds to a reduction of the temperature of the air passing the air coolers. The temperature of the airflow is then again increased as the air warms up due to the different loads applied to the freeze storage. The exergy input to the condenser and air cooler fans is lost for 100%. About 28% of the exergy input to the system are usefully applied to remove the cooling load from the frozen storage.

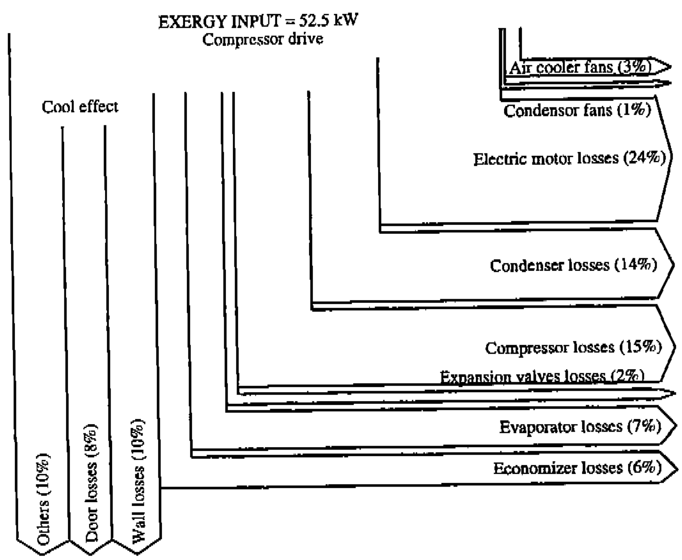

Fig. 1. - Exergy loss distribution in frozen storage.

Fig. 1 shows how the exergy input (=energy input) is lost or contributes to the cooling effect. The cooling effect is relatively large: 28% of the exergy input. The largest contributions to the exergy loss of such a system are:

- Losses of electric motor of compressor drive (24%)
- Compressor losses (15%)
- Condenser losses (14%)
- Wall losses (10%)
- Door losses (8%).

The refrigeration plant has been designed so that there is enough power available to cool down 25 ton of product from -17°C to -23°C twice a day. The electric motor has been selected for such operating conditions while for the conditions studied here the product is brought in with the final temperature. The result is that the electric motor operates at 60% of the nominal power and shows a relatively poor efficiency (η_{elec} is ca. 0.75).

Exergy analysis CO_2 transcritical

Substitution of the R22 refrigerant by CO_2 requires some modifications of the components. First of all the compressor size must be reduced. The cylinder volume of the reciprocating compressor has been reduced by a factor 5 so that the refrigerating capacity remains similar. Also the number of cylinders in de low stage has been reduced and in the high stage increased. Since the heat to be removed in the high-pressure side heat exchanger ("condenser") is relatively large, also the heat exchanger size has to be increased. For the simulations both the airflow rate and the heat transfer area have been increased with a factor 2. Also the expansion valve characteristics have been modified for CO_2 operation. For the same operating conditions as for the R22 plant, the electric motor showed a perfect fitting with the compressor load. Since the plant should be capable of cooling down 25 ton of product from -17°C to -23°C twice a day, a larger electric motor had also to be selected which again operates at about 60% of the nominal power.

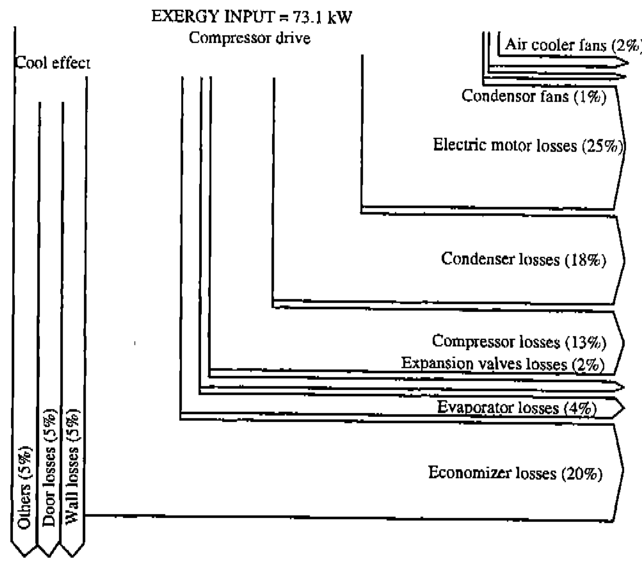

Fig. 2. - Exergy loss distribution in frozen storage (CO_2 transcritical cycle).

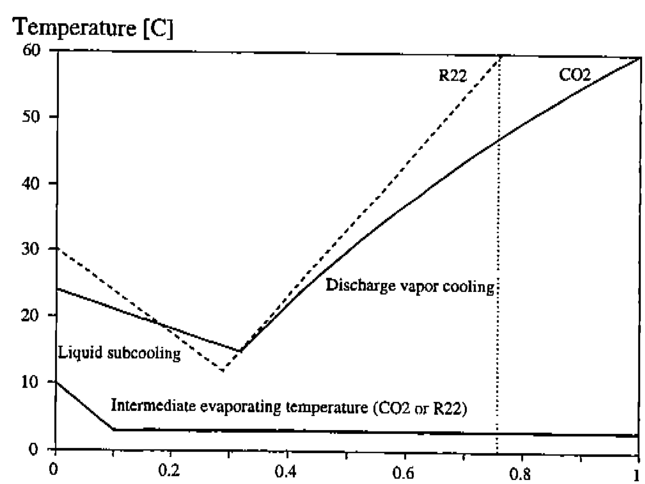

Fig. 3. - Q, T diagram of economizer.

Fig. 2 shows the results of the simulation. Now only 15% of the (larger) exergy input are usefully applied to remove the cooling load. This means that the CO_2 plant requires about 40% more exergy (=energy) input than the R22 plant. Fig. 2 shows further that the increased exergy loss contributions are specially originated in the "condenser" and economizer.

Fig. 3 shows a Q,T diagram of the economizer based on the heat exchanged in the CO_2 cycle. Also the lines for the original system (R22) with a smaller heat-exchanging ratio are shown. In a Q,T diagram of a heat exchanger the area between the temperature lines of the two streams gives a graphical representation of the exergy losses. From fig. 3 it appears that the exergy loss increase is mainly caused by the large heat load contribution of the low stage compressor discharge gas.

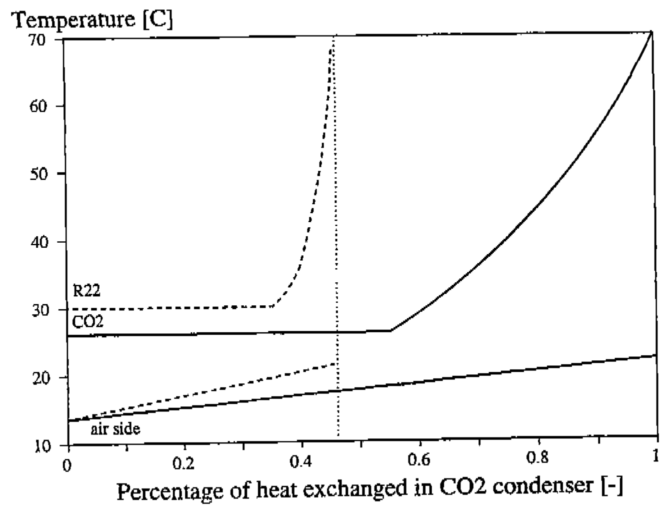

Fig. 4. - Q, T diagram van condenser.

Fig. 4 is a Q,T diagram for the condenser, again with the heat load in the CO_2 cycle as a reference. Since the vapor specific heat of CO_2 close to the critical point attains large values, the heat exchanged in the CO_2 cycle is twice as large as in the R22 cycle. Even though the condenser surface has been taken twice as large, the exergy losses are larger than for the R22 system.

The exergy losses in the CO_2 direct cycle can only be brought to the R22 system values with large efforts in reducing the temperature driving forces in the heat exchangers.

Exergy analysis CO_2 as secondary medium

When CO_2 is used as a secondary medium, keeping the R22 cycle as primary cycle, then the R22 will evaporate in the CO_2 condenser while CO_2 evaporates in the air coolers. Mainly two extra components are added to the system: the evaporator/condenser and the CO_2 pump. The same cooling capacity must now be delivered at a lower temperature so that the compressor size must be increased.

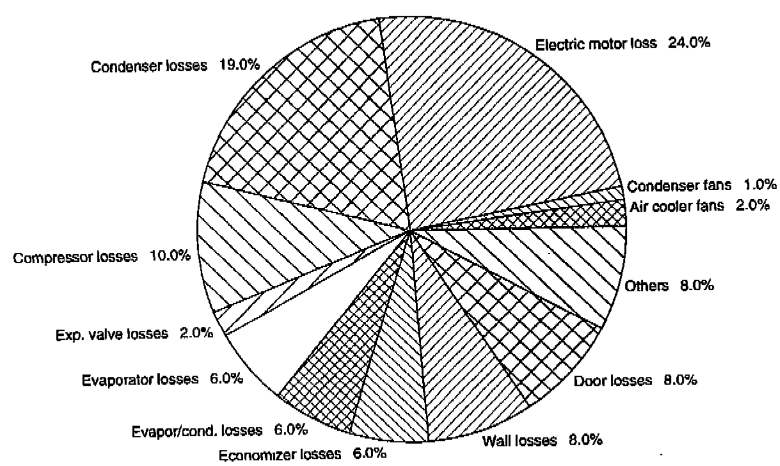

Fig. 5. - Exergy loss distribution in frozen storage (CO_2 indirect cycle). Exergy input = 56.2 kW.

Fig. 5 shows the relative contribution of the different components to the total exergy destruction of the system. The contribution of the condenser exergy destruction is now even larger than for the two previous options. Since the pressure ratio to be sustained by the compressor has increased, the end compressor temperature is larger and a larger quantity of heat must be exchanged in the condenser.

The plant requires for this system about 7% more exergy (=energy) input than for the R22 plant.

FRUIT STORAGE

Reference application

The complex consists of 10 chilled storage rooms for fruit, each one equipped with an 81 m^2 air cooler with three fans. The refrigeration plant consists of a four cylinder reciprocating compressor with a cooling capacity of 70 kW. The compressor delivery is controlled by a valve-lift-mechanism (4-2-0 cylinders). The refrigerant is HCFC 22. The condenser is an air-cooled condenser with 102 m^2 external surface. The condensation pressure is kept at 35°C or higher temperatures with a condensation pressure control.
The fruit is cooled down in September; by that time 184 ton fruit are chilled from field temperature (18°C) to storage temperature (-1°C) each 24 hours. The fruit is stored during the winter months; 950 ton fruit are then kept in the storage. Each storage room has a volume of 400 m^3. The air cooler fans remain continuously in operation even when the refrigerating plant is off. In this way homogeneous conditions can be guaranteed in the room.
The defrosting cycle has been included in the simulation but is has a negligible (storage) or small (cool down period) contribution to the exergy losses of the system.

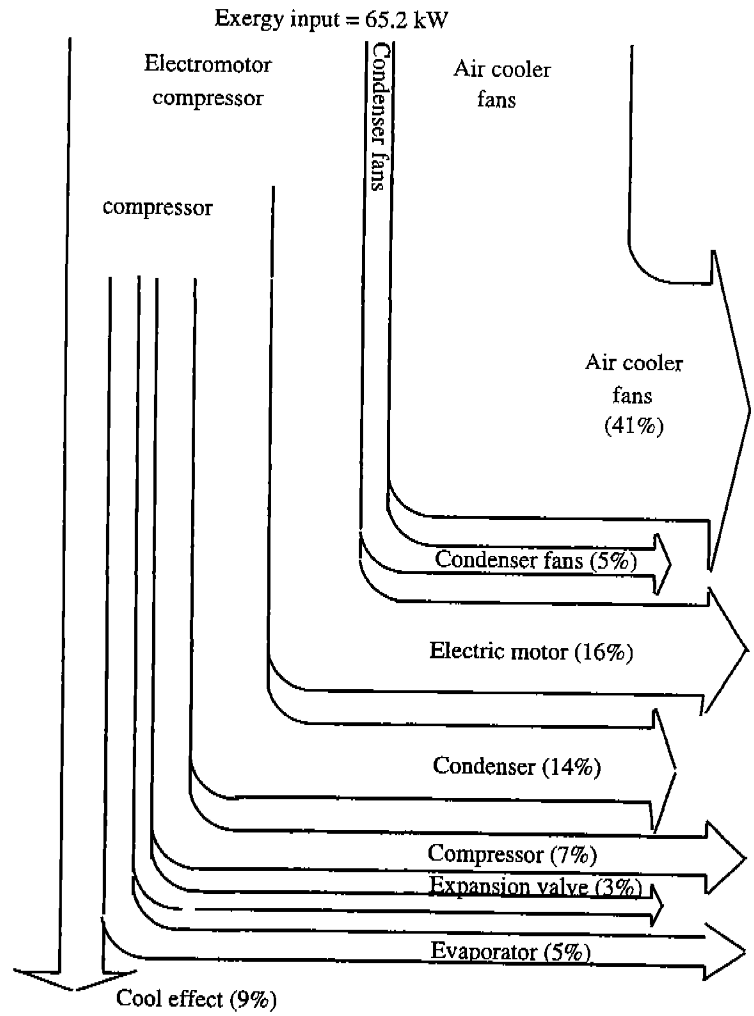

Fig. 6. - Exergy loss distribution in fruit storage.

Exergy analysis R22

Fig. 6 shows how the exergy input to the system is transformed in losses and cooling effect during the storage period. The percentages shown are averaged values for a 24 hours period. The cooling effect is here only 9% of the total exergy input. The largest losses are:

- Air cooler fans (41%)
- Losses of electric motor of compressor drive (16%)
- Condenser losses (14%)
- Compressor losses (7%)
- Condenser fans (5%).

Conservation of the fruit quality requires continuous ventilation. Most of the exergy loss is due to this requirement. The contribution of the refrigeration plant itself is relatively small in this case. Also in this application, the electric motor is operating under part load so that its efficiency is relatively low. The contributions of the cycle components are small except for the air cooled condenser - a result of the condensation temperature control.
Table 1 shows how the total exergy input is lost in the different components of the system during the cool down period.

Table 1. - Exergy loss distribution during cool down period in fruit storage. Exergy input = 258.4 kW.

	Percentage loss of total exergy input [%]
Air cooler fans	11
Condenser fans	10
Compressor drive	6
Condenser	33
Compressor	9
Expansion device	6
Evaporator	13
Cooling effect	14

CO_2 systems

These types of plants operate most of the time under storage conditions. For these conditions the refrigeration cycle contribution is relatively small. When CO_2 should be used instead of R22 then the exergy losses should increase but not as much as for a frozen storage application.

DISCUSSION / CONCLUSIONS

Frozen storage applications are responsible for about 60% of the energy consumption of the companies participating in the Dutch Cold Storage Association. An exergy analysis of a representative plant allows for an identification of the major causes of irreversibility in such plants. Due to the design strategy, the largest losses take place in the electric motor drive; a relative contribution which is not dependent of the refrigerant choice. The major exergy destruction takes place, in the R22 cycle, in the compressor and condenser. In the CO_2 cycle the economizer losses are also substantial. The compressor has been assumed to have a pressure ratio dependent isentropic efficiency, being ca. 70% for the simulated conditions. In practice compressors will show a lower efficiency so that the compressor contribution becomes even larger. Especially the high specific heat of CO_2 around the critical point causes a large increase of the exergy destruction in condenser and economizer. Low superheat and increased compressor isentropic efficiency are then required to keep the superheating contribution small.

Indirect systems seem to have a larger potential to reach exergy efficiencies comparable with the efficiencies of the existing systems. For these systems the condenser and condensing pressure control design should be the subject of optimization. Followed by an improvement of the compressor efficiency.

Fruit storage applications represent a much smaller contribution to the energy consumption of the sector and have been discussed in a lesser detail. The main cause of exergy destruction is here the storage ventilation followed by the electric motor drive losses. Both are refrigeration cycle independent so that it can be expected that the CO_2 cycles would not have a too large effect on the total exergy destruction rate.

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