Combined Industrial Cooling and Heating with Transcritical CO_2 Heat Pumps Utilising the Work of Expansion

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1. Abstract

The use of CO_2 as a refrigerant in transcritical vapour compression cycles has significant advantages, for systems which require simultaneous heating and cooling at approximately equal rates. However, then need for a compressor, to operate across high pressure differences, and the large throttle losses associated with these pressure differences have limited its use.

This paper describes a study carried out to evaluate the efficiency gains and cost benefits possible from such a system when a twin screw machine is used to both compress and expand the working fluid in a single unit. It also shows the values of the critical design parameters required to optimise the system's potential advantages when used in larger combined heating and cooling systems in industrial process and heat generation plants.

The results show that recovery of work from the expansion process improves the COP by 15 to 20%. For the design conditions specified in this paper, this implies that the expander is worth fitting if it can be installed for a cost of less than approximately €750/kW of shaft power input. Thus, depending on the operating conditions, transcritical CO_2 heat pumps using a compressor-expander can produce hot water at 90^{o}C with a COP of approximately 7, with thermal outputs of up to 1.5 MW. This could be extended with simple control strategies up to outputs of 10 MW.

2. Introduction

In many food industries, cooling and heating are required simultaneously in approximately equal quantities, e.g. for heating and cooling in pasteurisation units and production of hot water for CIP and cleaning duties. However, when using conventional vapour compression heat pumps to produce these two effects, only the heat withdrawn from the condenser during desuperheating is at an acceptable temperature level for the heating functions required and this is limited to some 5% to 15% of the cooling capacity, while the remaining heat transferred can only be used at temperatures below about 50°C if the heat pump is to have an acceptable COP. Fig 1 is a diagram of the essential features of a transcritical CO_2 cycle heat pump (TC-HP) with an internal heat exchanger. If this is used for such functions, there are significant advantages to be gained.

Firstly, constant pressure heat rejection in transcritical heat pumps differs from that in conventional systems in that none of the heat is transferred at constant temperature, as in the condensation process. Thus heat rejection in a transcritical CO_2 gas cooler can be effected in a counter flow heat exchanger with close temperature matching between the working fluid and the cooling water throughout the process, as shown by the change of state points 2 to 3 in Fig 2. By this means and applying a proper pinch point analysis, the cooling water can be heated to exit temperatures of up to 150°C. In this case, although the heat transfer coefficient of CO_2 varies locally in the transcritical region, it has been shown by Pitla et al [2001] that its average value is comparable to that of condensing refrigerants.

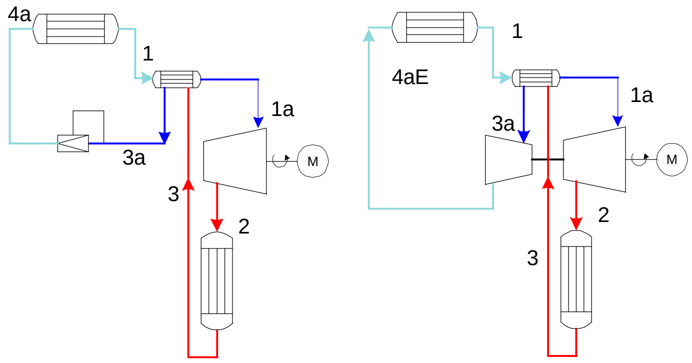

***Figure 1**: TC-HP with internal heat exchanger. Left: Conventional expansion and optimum gas cooler pressure control valve. Right: Compressor-Expander TC-HP with utilisation of the expansion work*

Secondly, as shown in states 3/3a to 4E/4aE in Fig 2, due to expansion occurring in the transcritical region, the power recoverable from this process is much larger than in conventional vapour compression cycles. Hence, provided that an efficient expander is available for the admission of supercritical fluid and the expulsion of a two-phase fluid, the potential for COP improvement by this means is higher [Robinson & Groll 1998, Huff, Radermacher & Preissner 2003].

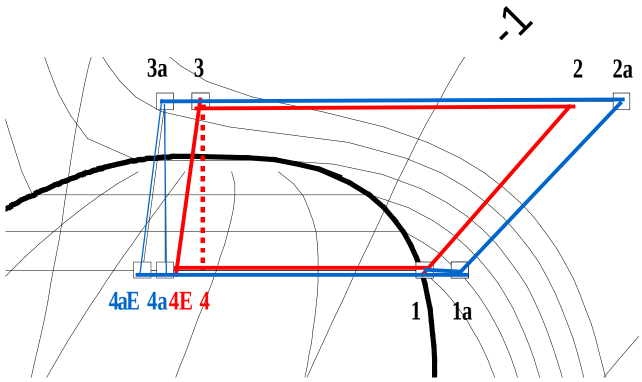

***Figure 2:** Transcritical CO_2 Heat Pump cycle states on Log p-h coordinates, showing the paths with and without utilisation of expansion work, with internal heat exchange (blue cycle) and without internal heat exchange (red cycle).*

Twin screw machines can be used both for compression and expansion. However, a serious limitation in their use is that they incur large bearing loads and hence this limits the pressure difference across which they can be operated successfully. It has, been shown [Stosic et al 2002] that by using the same machine both for compression and expansion, as shown in Fig 3, the rotor forces on the compressor section can be partially balanced by those on the expander section and by this means the combined compressor-expander should be able to fulfill both functions in a TC-HP.

Further advantages are associated with the reduced volume circulation rate per unit of heat transferred due to the high density of CO_2 at the required operating pressures and temperatures. These are well recognised in the refrigeration industry [Hansen et al. 2001 & 2003, Rolfsman 2003] and include reduced compressor size, pipe and valve dimensions.

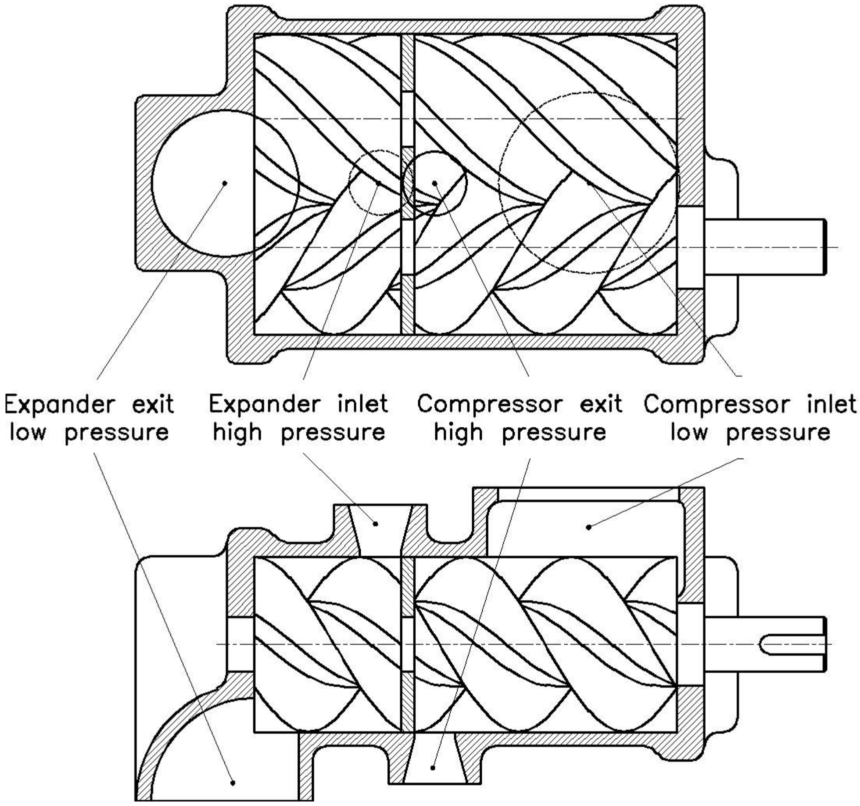

Figure 3: *Combined twin screw compressor-expander with balanced rotors.*

Two specific issues have to be resolved when evaluating the potential of a CO_2 TC-HP. These are that:

1. The optimum COP at a given gas cooler pressure depends on the evaporation temperature and CO_2 gas outlet temperature.
2. The available heat sources and the inlet water temperature have great influence the optimum operating point and the COP.

The following study was carried out to determine what advantages could be obtained from the use of a TC-HP fitted with a balanced rotor screw compressor-expander, given the need to produce hot water at 80^{o}C.

3. Transcritical CO_2 Heat Pump Analysis

3.1 Available Heat Sources

As for all heat pumps, the COP is improved if the evaporator temperature can be maximised. Figure 4 shows the calculated values of the COP for a TC-HP, with and without power recovery from an expander, compared to a water vapour heat pump using 3 stages with an intercooler, to produce 80°C water assuming various evaporation temperatures. The “R744 TC-HP” configuration is for a traditional cycle with optimum gas cooler control and expansion valve.

The “R744 Compressor-Expander TC-HP” configuration is calculated assuming that an expander with a constant 70% isentropic efficiency is included. At higher evaporation temperatures the Compressor-Expander TC-HP COP exceeds even that of the water vapour alternative (R718 HP) by 15% to 25%. The advantages of recovering work from the expansion process and operating at high evaporation temperatures are thus clearly demonstrated.

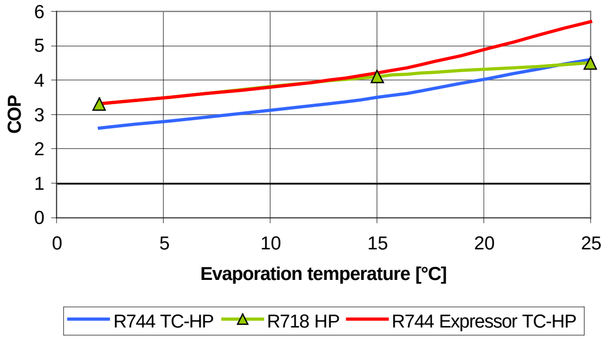

Figure 4*: TC-HP COP dependency of evaporation temperature with fixed gas cooler temperature of 45°C and optimum gas cooler pressure control.*

The following possible heat sources for a TC-HP are listed in order of preferred evaporator temperature level.

1. Waste heat source from exhaust gas cooling and condensation
2. Waste heat from processes
3. Condensation heat from other refrigeration units (20°C - 30°C)
4. Geothermal source: Acceptable COP compared to 20°C evaporation. Prohibited in some areas, higher investment costs.
5. Sewage and ambient water at 7°C: Reduced COP compared to 20°C evaporation.
6. Ground coil: Reduced COP during winter operation. Reduced COP compared to 20°C evaporation. Higher investment costs.
7. Air: Significant reduction of COP during winter operation with an evaporation design temperature down to –20°C.

When using air as the heat source, a more complex control system is needed for the heat pump to operate with an optimum COP, since the evaporation temperature varies seasonally.

3.2 Internal Heat Exchanger

At high evaporation temperatures, an internal heat exchanger (IHX) is required to enable the heating system to produce water at the required temperature, which in this case is 80°C. The dependence of the gas cooler inlet temperature on the IHX effectiveness is shown in Figure 4 for a TC-HP fitted with a compressor-expander in which both the compression and expansion isentropic efficiencies are 70% and the evaporation temperatures are taken as 0°C, 16°C and 20°C, respectively.

It can be seen that for an evaporation temperature of 20°C, an IHX effectiveness of 0.4 is needed to obtain a gas cooler inlet gas temperature of 90°C. This results in a gas cooler outlet temperature difference of only 10°C if hot water is to be produced at 80°C by counterflow heat exchange.

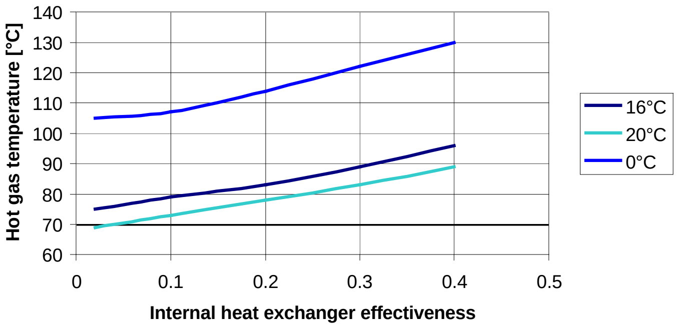

Figure 4: *Gas cooler inlet temperature as a function of the internal heat exchanger effectiveness in a TC-HP fitted with a compressor-expander working at different evaporation temperatures. The gas cooler pressure is assumed to be 105 bar and the gas cooler exit temperature is at 45°C.*

Although increasing the IHX effectiveness permits more heat to be delivered to the hot water, its effect on the COP is not favourable since the compression work required is thus increased while the recoverable adiabatic work of expansion is simultaneously decreased. This is due to the divergence of the constant entropy lines as the temperature increases, as shown, on p-h coordinates, in Figure 2. Thus, the heat delivered to the hot water is increased by the enthalpy gain from 2a-2 but the enthalpy recovery due to expansion work from 4a-4aE<4-4E and the enthalpy input due to compression work is increased since 1a-2a<1-2. The overall effect of this is that by increasing the evaporation temperature between 0°C and 20°C, the ideal COP is decreased by approximately 10%.

Thus raising the evaporation temperature to increase the COP has the drawback that an IHX is needed to meet the demand for 80°C hot water.

<u>3.3 Optimum Gas Cooler Pressure</u>
For a TC-HP an optimum gas cooler pressure exists where the COP has a maximum value. The optimum gas cooler pressure depends on two parameters, namely the gas cooler exit temperature and the evaporation temperature.

As shown in Figure 5, where the gas cooler exit temperature is fixed at 45°C, the optimum gas cooler pressure is hardly affected by the evaporation temperature in the range from 0°C and 20°C. Based on these results, the setting of the gas cooler pressure at a constant value of approx. 105 bar would provide an adequate design value.

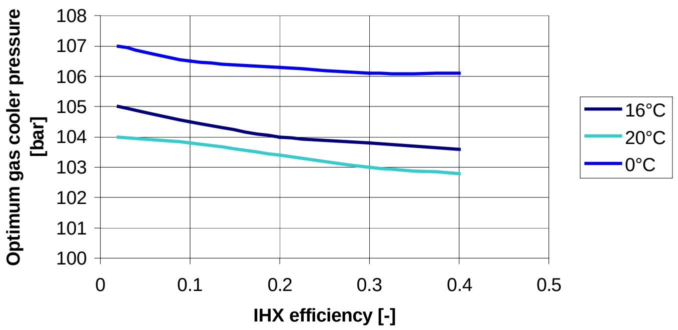

Figure 5: *Optimum gas cooler pressure at different evaporation temperatures and IHX efficiency. Gas cooler temperature fixed at 45°C.*

The gas cooler exit temperature is governed by the inlet water temperature and has more influence on the optimum pressure. The gas cooler exit temperature should be as low as possible in order to obtain a high COP. Lowering the gas cooler exit temperature reduces the available heat that can be transferred to the compressor suction side to provide 80°C hot water. Hence, the IHX effectiveness must be increased to obtain the required superheat, as shown on the right hand axis in figure 6.

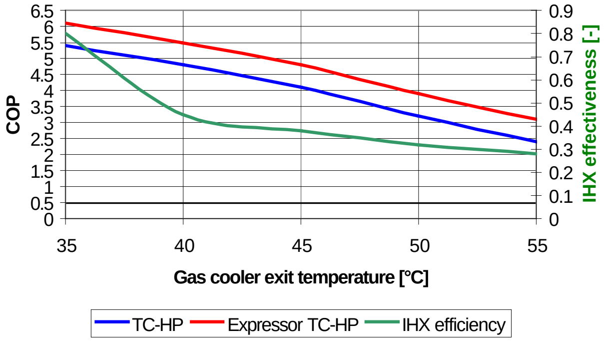

Figure 6: *COP dependence on the gas cooler exit temperature at fixed evaporation temperature (20°C) and fixed gas cooler pressure (105 bar). Right axis: Required IHX efficiency for 80°C hot water production.*

In Figure 7, the COP and the optimum gas cooler pressure are shown at varying gas cooler exit temperatures.

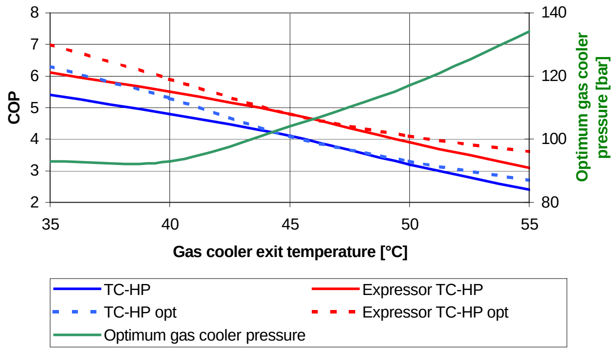

Figure 7: *Maximum COP at optimum gas cooler pressure (right axis) at fixed evaporation temperature (20°C)*

The dotted lines in the figure show both the optimum COP and the COP at a fixed gas cooler pressure of 105 bar. In the case of 35°C gas cooler exit it is not possible to maximise the COP by controlling the gas cooler pressure and at the same time produce 80°C water. Therefore, in the figure the lowest viable operational gas cooler pressure at 20°C evaporation is limited to 93 bar.

Figure 8 shows that by operating the HP at a constant pressure of 105 bar, there is a loss of less than 7% in the COP, for a deviation of ±5K from the design gas cooler exit temperature of 45°C, compared to that obtainable by operating it at a controlled optimum gas cooler pressure.

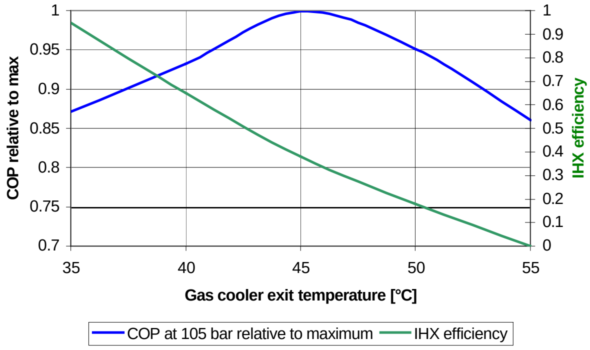

Figure 8: *COP with operation at fixed gas cooler pressure 105 bar relative to COP at the optimum fixed evaporation temperature (20°C).*

4. TC-HP Design Specification

Based on the above results, the following design specification was selected to evaluate the performance of a TC-HP with balanced rotor, 4 port screw compressor-expander device.

Design pressure (gas cooler/evaporation): 105 bar/57 bar
Evaporation temperature: 20°C
Gas cooler exit temperature: 45°C
Hot water production temperature: 80°C
IHX efficiency: 0.4

Table 1: Calculated performance of 4 port, balanced rotor screw compressor-expander.

Suction volume	M^3/hr	150	175	200
Expected performance without/with bearing losses				
Compressor power consumption	KW	219/243	244/271	281/312
Expander power	KW	58/53	70/64	80/73
η_{vol} of compressor	%	80	84	84
η_{is} of compressor	%	65	69	69
η_{is} of expander	%	66	68	68

5. Savings Provided by a Compressor-Expander TC-HP

For the given design specification, the savings provided by a TC-HP have been evaluated both for CHP production and in dairy installations.

5.1 CHP systems

An evaluation of the feasibility and cost attractiveness of industrial heat pumps in heating systems requires consideration of such parameters as electricity cost, share of non-controllable electricity production, share of fossil fuel sources, taxation rates on fossil fuels, assessable mediate temperature sources for heat uptake, and existing heat distribution systems. Figure 9 shows the yearly operational and maintenance savings provided by a Compressor-Expander TC-HP for different sized CHP systems, when applied to conditions in Denmark (2003, 7.5 DKK to €1). It follows that a separate analysis, based on local governing conditions, must be carried out for each individual country to determine the savings possible by applying a Compressor-Expander TC-HP in a gas engine equipped CHP plant elsewhere than Denmark.

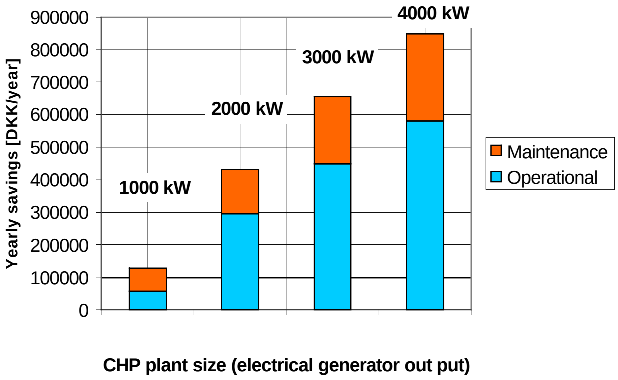

Figure 9: *Yearly operational and maintenance savings provided by an Compressor-Expander TC-HP for CHP systems with 4,500 annual operation hours*

5.2 Dairy industry

A cost benefit has been carried out assuming Danish industrial gas prices. In this case, the heat pump facilitates gas savings corresponding to the heat production requirement. The operating cost of the heat pump corresponds to the difference in energy consumption of the CO_2 heat pump compared to an ammonia chiller serving an ice bank. The COP of an ammonia chiller operating at -5°C/35°C is 4.2 compared to approx. 3.0 of a CO_2 heat pump at -5°C/90°C at 105 bar gas cooler pressure. The key figures and savings provided by the CO_2 heat pump are summarised in table 2:

Table 2: Savings provided by a Compressor-Expander TC-HP in the dairy industry

Cooling	Gas consump-tion, yearly	Gas cost, Yearly	Electrical cost CO_2, yearly	Electrical cost NH_3, yearly	Total savings, yearly	
KW	Nm³	DKK	DKK	DKK	DKK	€
400	202105	404211	192000	137143	349353	46580
600	303158	606316	288000	205714	524030	69871
800	404211	808421	384000	274286	698707	93161
1000	505263	1010526	480000	342857	873383	116451

6. Conclusions

Both transcritical CO_2 heat pumps with and without a Compressor-Expander offer attractive benefits for specific industrial applications such as pasturisations units or small scale district heating systems. Based on the analysis, transcritical CO_2 Compressor-Expander heat pumps are the best present available and offer sustainable technology for heating systems up to 1.5 MW heat production, and potentially up to 10 MW even with simple control strategies. Dependent on the possible operating conditions, a Compressor-Expander TC-HP may produce 90°C hot water with a COP of approx 7.

Utilisation of the work of expansion leads to a 15–20% improvement in the COP. In economic terms, the break even cost of an expansion device meeting the design specifications in this paper is approx. €40,000.

The TC-HP may be installed as stand alone unit in existing equipment or as replacement of the evaporative condenser in conventional industrial installation. Based on the calculated savings, it is plausible in the future that new refrigeration system designs with a double $CO_2/NH_3/CO_2$ cascade may become highly attractive.

7. Nomenclature

COP:	Coefficient of Performance	η_{is}:	Isentropic efficiency [-]
IHX:	Internal Heat Exchanger	η_{vol}:	Volumetric efficiency [-]
TC-HP:	Transcritical Heat Pump		

8. References

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