

# A comparison of the energy consumption for CO<sub>2</sub> compression process alternatives

S Jackson and E Brodal

Institute for Engineering & Safety, UiT Norges arktiske universitet, Norway  
Email: [steve.jackson@uit.no](mailto:steve.jackson@uit.no)

**Abstract.** The efficient transportation of large volumes of CO<sub>2</sub> generally requires pipelines that will operate above the critical pressure of CO<sub>2</sub>. Since most capture processes release CO<sub>2</sub> at low pressure, compression of CO<sub>2</sub> from the point of capture to pipeline will generally be required. The compression duty can be achieved using *conventional* multi-stage compressors or using newer shockwave type compressors. Pumping could also be used if CO<sub>2</sub> is condensed below its critical point. This paper presents a comparison the energy consumption associated with these compression process alternatives. The focus of the review is on the clarity of the comparison and the careful optimisation of each of the scheme considered. The main finding is that the performance advantages claimed for improved CO<sub>2</sub> compression process schemes are often optimistic because the based-line scheme compared against is not well optimized.

## 1. Introduction

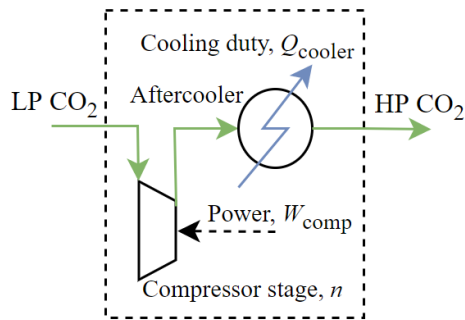
The addition of carbon capture to a power station has a significant impact on efficiency. For a typical coal-fired plant with post combustion capture, the net efficiency could be expected to drop by as much as 7 to 12% due to the heat and power requirements of the capture process [1, 2].

Expressing the efficiency penalty in terms of equivalent lost electrical power production per unit CO<sub>2</sub> captured (kWh<sub>e</sub>/tCO<sub>2</sub>) is convenient for assessment purposes and this definition shall be used in this paper. Lucquiaud et al. [3] report that for an “nth of a kind CCS plant with current state-of-the-art solvent technology” as 250-300 kWh<sub>e</sub>/tCO<sub>2</sub>. Compression of CO<sub>2</sub> is expected to consume 90-120 kWh<sub>e</sub>/tCO<sub>2</sub> of electrical power [4], making the contribution 30 to 50% of the total. As advances are made in capture technology, the importance of the energy consumption of the compression process will only become more important.

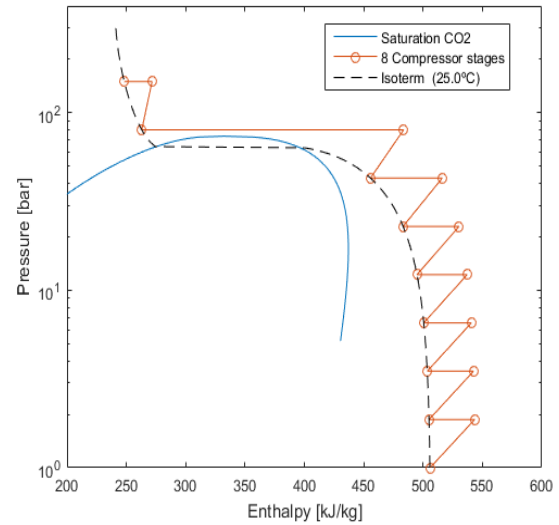
Multi-stage integrally geared type compressors have been used in high-pressure CO<sub>2</sub> applications for decades [5] and are identified in this article as the *conventional* approach to CO<sub>2</sub> compression. This type of compressor uses intercooling between each stage and typically has a pressure ratio around two [6]. Little useful heat is generated during compression and waste heat is normally rejected to a cooling utility.

A *conventional* eight-stage CO<sub>2</sub> compression process with pumping from above critical pressure is illustrated in Figure 1 and Figure 2. In this model, the compressors work isentropic,  $\eta_{is} = 1$ , and with a constant pressure ratio,  $r_g = 1.87$ .





**Figure 1.** Illustration of a *conventional* compression process with  $n$  stages.

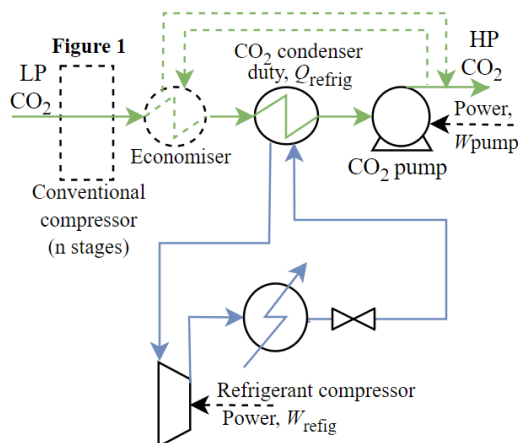


**Figure 2.** Illustration of a *conventional* eight-stage compressor process in a pressure-enthalpy diagram.

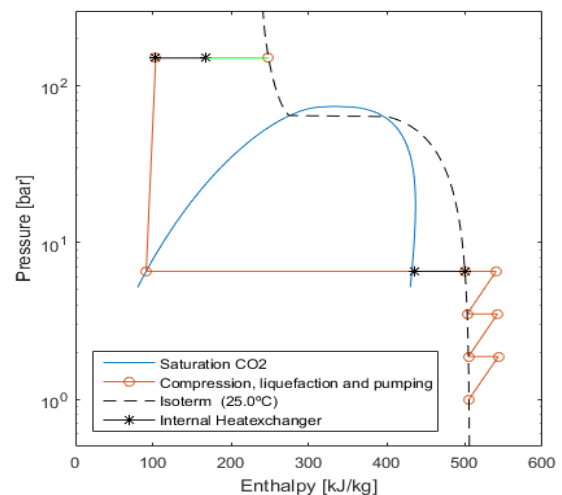
Above its critical pressure (7.38 MPa), CO<sub>2</sub> will behave with properties similar to that of a liquid at 25°C and in an eight-stage compressor process the final stage of compression can be replaced with a pump. To enable a process where CO<sub>2</sub> is pumped to the final pressure from below its critical pressure, refrigeration is needed to cool and condense CO<sub>2</sub> below ambient temperature.

Several studies have investigated compression schemes that utilize pumping of CO<sub>2</sub> from below its critical pressure where an external refrigerant is used to condense below ambient temperature CO<sub>2</sub> [7, 8]. Again, these studies have presented optimized schemes that demonstrate potentially significant CO<sub>2</sub> compressor power savings.

Figure 3 and Figure 4 show an illustration of a three-stage compression process with refrigeration and pumping, where the compression process uses isentropic compressors with pressure ratio,  $r_8 = 1.87$ .



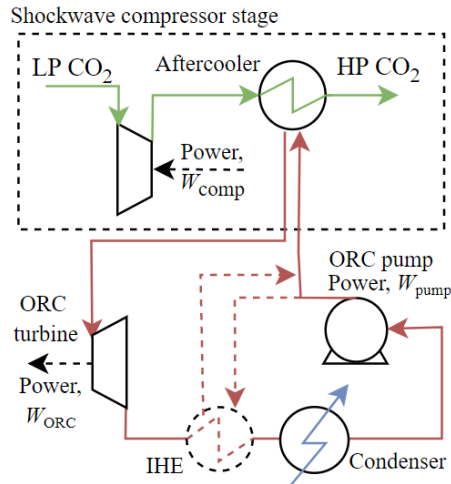
**Figure 3.** Illustration of a *conventional* compression process with refrigeration and pumping from below critical pressure.



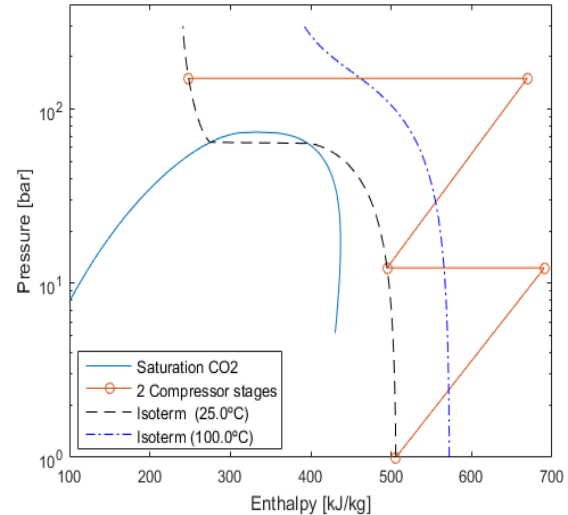
**Figure 4.** Process with three compressors ( $r=1.87$ ), internal heat exchanger, refrigeration unit and pump.

Several studies have looked at the opportunity to utilize wasted heat from CO<sub>2</sub> compressor aftercoolers. Although some research has been conducted into the potential benefits of this approach when applied to *conventional*, low pressure ratio compressor designs [9], the focus of research activity has been *unconventional* compressor designs with high pressure ratios [10, 6, 11].

Figure 5 and Figure 6 illustrate a two-stage compression process where heat is recovered from the compressor aftercoolers in an ORC cycle. The ORC cycle is shown with an optional Internal Heat Exchanger (IHE), sometimes referred to as a recuperator.



**Figure 5.** Illustration of a single-stage compression processing with heat recovery using an ORC.



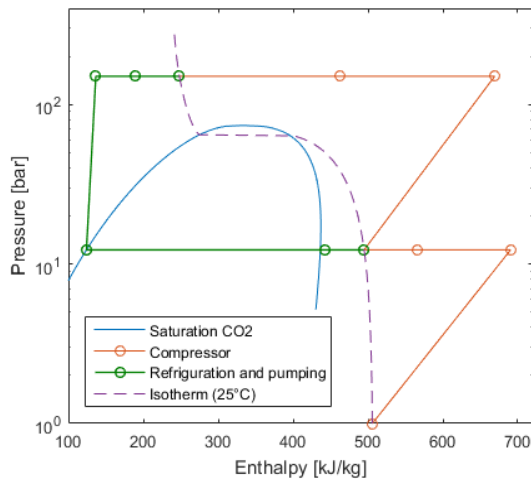
**Figure 6.** Two Compressor Stages ( $n = 2$ ) and Pressure Ratio  $r_2 = 12$ .

Studies into heat recovery from CO<sub>2</sub> compressor aftercoolers have also generally found that a reduction in overall power is achievable. A recent focus for research and development work has been a shockwave type compression technology under developed [12]. This technology claims to offer several advantages over a *conventional* compressor design including “reduced capital costs, smaller footprint, and reduced parasitic plant impact” [13].

A less studied alternative of a pumped processes is one where the heat from compression is to run an absorption refrigeration process. If it is assumed that a bleed before the last compression stage is cooled down using absorption refrigeration process that is driven by the heat from the compressor, as shown in Figure 7, the amount of CO<sub>2</sub> which is liquefied is limited by the amount of useful heat generated in the compression stages and the efficiency of the absorption refrigeration process.

The comparison of the performance of different CO<sub>2</sub> compression schemes for real systems depends on a large number of parameters. For example, compressor efficiencies (per stage), equipment pressure losses, exchanger temperature approaches, etc., all need to be set on a consistent and realistic basis for a valid comparison. None of the studies mentioned have compared the full range of alternative CO<sub>2</sub> compression approaches listed above on a common basis and there is little consistency in the approach used to accounting for the relative value of waste heat and refrigeration duties.

The aim of this study is, therefore, to review all the main process alternatives for CO<sub>2</sub> compression on a simple and consistent basis so that a comparison can be made of the relative performance of these different approaches to CO<sub>2</sub> compression.



**Figure 7.** Illustration of a two-stage compression process where absorption refrigeration used to liquefy a CO<sub>2</sub>-bleed using the heat generated in compression.

## 2. Method

The method adopted in this research work is to consider the review of the alternative compression processes in two parts: firstly, a theoretical review of minimum power consumption for a broad range of compressor system design cases based on the performance of idealized thermodynamic cycles; and subsequently, a review of the practically achievable minimum power consumption for a more focused selection of promising cases based on a set of assumed process equipment performance parameters. The first set of work is referred to in this report as *idealized modelling* the second *engineering design study*.

In both parts of the modelling work the feed stream to the compressor is consider to be pure CO<sub>2</sub>, the feed pressure is 1 bara and the pipeline pressure 150 bara. Other modelling parameters and the tools used to construct the process models vary between the *idealized* and *engineering* parts of the study and are described under the following headings.

### 2.1. Idealized modelling cases

The performance of each of the idealized processes is modelled in Matlab based on enthalpy data from the CoolProp fluid package. Coolprop uses the Span & Wagner EOS [14] to calculate CO<sub>2</sub> properties which is implemented in NIST REFPROP and generally considered to be the benchmark for pure CO<sub>2</sub> property prediction.

- All machinery operates ideally with isentropic efficiency,  $\eta_{is} = 1$
- Waste heat is rejected at a minimum temperature,  $T_C = 25^\circ\text{C}$
- Heat exchangers have a temperature approach,  $\Delta T_{min} = 0^\circ\text{C}$  and pressure drop,  $\Delta P = 0$  bar
- Compressors stage pressure ratio,  $r$ , is constant, i.e.,  $r_n = 150^{1/n}$ , where  $n$  is the number of stages

**2.1.1. Conventional compression with refrigeration and pumping.** In a *conventional* CO<sub>2</sub> compression processes the pressure ratio per-stage,  $r$ , is usually limited to around 2.0 [6] and based on this, the minimum number of stages for these cases is set to 8, giving  $r_8 = 1.87$ . The total compressor power,  $W_{comp}$ , is the sum of the power for each stage,  $W_n$ :

$$W_{comp} = \sum_{n=1}^n W_n \quad (1)$$

When CO<sub>2</sub> is condensed above its critical pressure the refrigeration cooling duty,  $Q_{refrig}$ , and the refrigeration compressor power,  $W_{refrig}$ , reduces to zero. In all other cases, the refrigeration unit duty consumption is assumed to be the same as an ideal Carnot cycle, which represents the highest efficiency possible between two reservoirs with temperatures  $T_H$  and  $T_L$ .

$$\text{COP}_{refrig}^{ideal} = \frac{Q_{refrig}}{W_{refrig}} = \frac{T_L}{T_H - T_L} \quad (2)$$

The power consumption,  $W_o$  for refrigeration and pumping models, where the  $\text{CO}_2$  is condensed below its critical pressure is the sum of  $\text{CO}_2$  compression duty,  $W_{\text{comp}}$ , refrigeration duty,  $W_{\text{refrig}}$  and pumping duty,  $W_{\text{pump}}$ .

$$W_o = W_{\text{comp}} + W_{\text{refrig}} + W_{\text{pump}} \quad (3)$$

If the  $\text{CO}_2$  exiting the pump at pipeline pressure is cold, so an economizer is used reduce the temperature before the refrigeration process to minimize the refrigeration duty, see Figure 3.

$\text{CO}_2$  has a triple point of  $-56.6^\circ\text{C}$  and 510 kPa, so if freezing is to be avoided in the liquefaction process,  $\text{CO}_2$  must be condensed above  $-56.6^\circ\text{C}$ . In the *conventional* eight-stage compressor arrangement, the first opportunity to condense  $\text{CO}_2$  above its freezing point is after Stage 3 at 650 kPa (see Figure 4).

**2.1.2. Shockwave compression and heat recovery.** The  $\text{CO}_2$  compression is an energy intensive process with high stage outlet temperatures. Heat in the aftercoolers is available at a range of temperature levels. The total amount of heat available being:

$$Q_{\text{HR}} = \int_{25^\circ\text{C}}^{\infty} q_{\text{HR}} dT \quad (4)$$

The maximum efficiency for a heat engine working between two reservoirs with temperatures  $T_H$  and  $T_L$ , is the Carnot efficiency:

$$\eta_{\text{max}} = 1 - \frac{T_L}{T_H} \quad (5)$$

The amount of work that can theoretical be recovered from the heat above temperature  $T_{\text{HR}}$ , using an infinite number of Carnot cycles and ambient temperature cooling  $T_C$  is therefore:

$$W_{\text{HR}}^{\text{ideal}} = \int_{T_{\text{HR}}}^{\infty} w_{\text{HR}}^{\text{Carnot}} dT = \int_{T_{\text{HR}}}^{\infty} q_{\text{HR}} \left(1 - \frac{T_C}{T}\right) dT \quad (6)$$

The total power consumption of the shockwave compression with heat recovery cases is calculated based on the compression power minus the maximum work that can be recovered in the heat recovery process,  $W_{\text{HR}}^{\text{ideal}}$ , as illustrated below.

$$W_o = W_{\text{comp}} - W_{\text{HR}}^{\text{ideal}} \quad (7)$$

**2.1.3. Absorption refrigeration.** Using heat at high temperature ( $T_H$ ) for cooling at low temperature ( $T_L$ ), the theoretical maximum coefficient of performance for an ideal cycle is [15]:

$$\text{COP}_{\text{abs}}^{\text{ideal}} = \frac{Q_L}{Q_H} = \frac{T_L}{T_H} \cdot \frac{T_H - T_C}{T_C - T_L}, \quad (8)$$

where  $T_C$  is the ambient temperature, e.g. cooling water.

Figure 7 shows a process where a  $\text{CO}_2$  bleed stream is liquefied at  $T_L \approx -40^\circ\text{C}$ . This absorption refrigeration process will have  $\text{COP}_{\text{abs}}^{\text{ideal}} = 0.83$ , assuming  $T_C = 20^\circ\text{C}$  and that all heat above  $T_H = 100^\circ\text{C}$  can be used in the refrigeration process.

A review of the literature for absorption refrigeration shows that the ideal COP is far from realistic when considering real systems. For example, the ideal coefficient of performance for an absorption refrigeration between  $T_H = 120^\circ\text{C}$  and  $T_C = 5^\circ\text{C}$  is  $\text{COP}_{\text{abs}}^{\text{ideal}} = 4$ , while a realistic achievable value is

$COP_{abs}^{real} \approx 1.0$  [16]. Because of this, the assessments made here consider a range of COP values from a maximum of  $COP_{abs}^{ideal}$  to a minimum of 0.4 (10% of the maximum,  $COP_{abs}^{ideal}$ ).

## 2.2. Engineering design study

The results from the *idealized* modelling work identify both *conventional* refrigeration and pumping schemes with between 4 and 6 stages and shockwave compression schemes with 1 to 3 stages as having lowest power consumption. Accordingly, these cases were studied in more detail.

Compressor efficiency is a key parameter and must be selected on a reasonable basis for a fair comparison to be made. This study uses  $\eta_{is} = 0.85$  for both *conventional* and shockwave type compressors based on performance claims made for shockwave compressors under development [17], indicating typical adiabatic efficiencies of 85% for both shockwave compressors with a pressure ratio of 9 and *conventional* centrifugal compressor stages operating with a pressure ratio of 2.

Other design parameters used in this part of the study work are summarized below.

- Isentropic efficiency for pumps and turbines,  $\eta_{is} = 0.85$
- Waste heat is rejected at a minimum temperature,  $T_{HR} = 25\text{ }^{\circ}\text{C}$
- Heat exchangers have a temperature approach,  $\Delta T_{min} = 5\text{ }^{\circ}\text{C}$  and pressure drop,  $\Delta P = 30\text{ kPa}$
- Maximum stage pressure ratio for *conventional* compressors,  $r = 2.0$
- Maximum stage pressure ratio for shockwave compressors,  $r = 10.0$
- Minimum  $\text{CO}_2$  pumping pressure =  $\text{CO}_2$  critical pressure + 10 bar = 8.39 MPa
- Minimum sub-cooling prior to pumping,  $\Delta T_{sub} = 10\text{ }^{\circ}\text{C}$

Aspen HYSYS was used to build the process models required and energy flows were calculated using the Peng Robinson (PR) equation of state. Studies have confirmed that PR generally provides reasonable accuracy in predicting the relevant properties for pure  $\text{CO}_2$  apart from region immediately around the critical point [18, 19].

Additional variables relevant to this part of the study work are the compressor stage pressure ratios; the ORC operating parameters; ORC working fluid and ORC process design. The selection of these parameters was the result of a set of optimization studies that are described under the following headings.

**2.2.1. Optimisation of conventional compression cases.**  $\text{CO}_2$  compressor power is minimized when the stage pressure ratio is allowed to vary. Overall power is also minimized when  $\text{CO}_2$  is pumped from immediately above its critical pressure. In the optimization of the *conventional* compressor cases the compressor stage pressure ratios were varied up to the 2.0 limit. Where this limit was reached in any single-stage, a new case was added to the optimization study where the pressure ratio limit was increased to 4.0 for the stage in question. In practical terms, this represents the addition of a new compressor wheel to the compressor design and not a new stage with after cooling. The  $\text{CO}_2$  was pumped from 8.39 MPa in all cases.

Ammonia is a refrigerant that exhibits good energy efficiency over the temperature ranges of interest to this study. However, when heat exchanger pressure drops are introduced into the compressor design, the required refrigeration temperature after three stages of compression falls below  $-33\text{ }^{\circ}\text{C}$  and an ammonia based system would be required to operate below atmospheric pressure. For this reason, the cases considered in this part of the study are for 4 and more stages of compression. In all cases a simple single-stage refrigeration cycle is used.

**2.2.2. Optimisation of shockwave compression and heat recovery cases.** In this study, heat is recovered using a single-stage ORC, mirroring the approach of Pei et al. [10] and Farajollahi et al. [9]. Selection of an optimum ORC working fluid and process scheme is a complex task in itself and in this study, and based on the work of Lai et al. [20] three working fluids and two process designs were identified for assessment: cyclo-pentane, n-pentane, n-butane, an ORC design with Internal Heat Exchanger (IHE) and an ORC design without IHE as illustrated in Figure 5.

In each of the ORC design cases the maximum pressure was set to 90% of the working fluid critical pressure and the cooling temperature set equal to that in the compression process. The remaining



ORC operating parameters and the CO<sub>2</sub> compressor stage pressure ratio were optimized for minimum overall specific power,  $W_o$ . Results for all cases are presented in part 3.

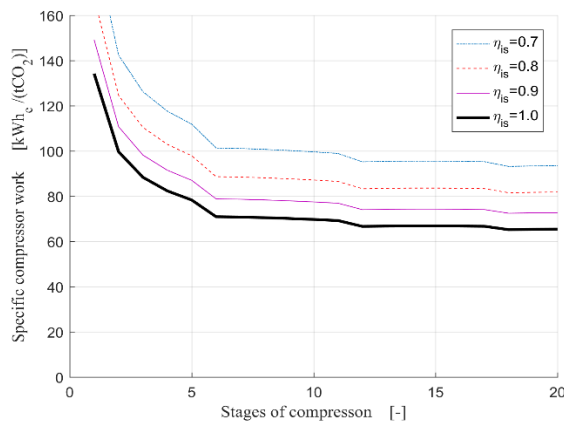
**2.2.3. Sensitivity cases.** To ensure that the selected process design parameters do not favor one of the cases studied over the others, a set of sensitivity studies for key process parameters was conducted. The convention and shockwave compression schemes with lowest overall specific power consumption were used as the basis for the study and the parameters adjusted were as follows:

- Compressor, turbine & pump efficiency,  $\eta_{is} = 0.75$  to 0.90
- Heat exchanger pressure drop,  $\Delta P = 10$  to 50 kPa
- Cooling temperature,  $T_c = 15$  to 45 °C

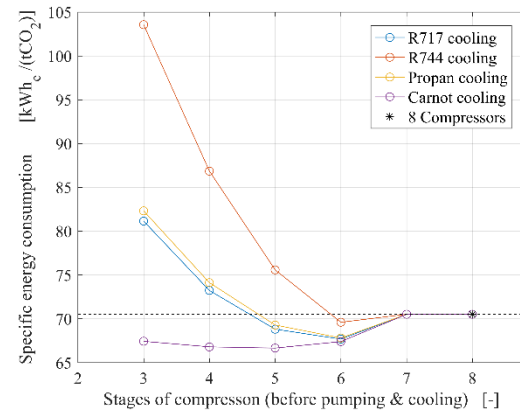
### 3. Results

#### 3.1. Idealized modelling

**3.1.1. Compression, liquefaction and pumping.** Figure 8 and Figure 9 show how the energy usage changes with the number of compressor stages used in the process.

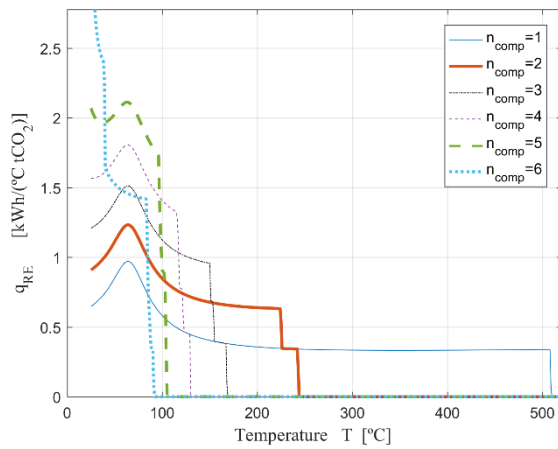


**Figure 8.** Impact of compressor stages on power for multi-stage compressor designs with equal pressure ratio and isentropic efficiency between 0.7 and 1.0.

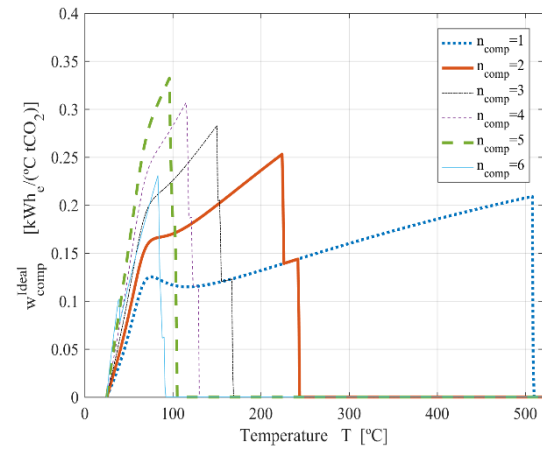


**Figure 9.** Energy costs in processes with  $n$  compressors ( $r = 1.87$  pressure ratios), pump and ideal one-stage refrigeration cycles.

**3.1.2. Compression with heat recovery.** The amount of heat available at different temperatures is illustrated in Figure 10, where the total energy above 25 °C. The maximum work  $W_{HR}^{ideal}$  is illustrated at different temperatures in Figure 11. Figure 12 shows the overall specific energy consumption of the compression process with heat recovery at three different temperature levels.

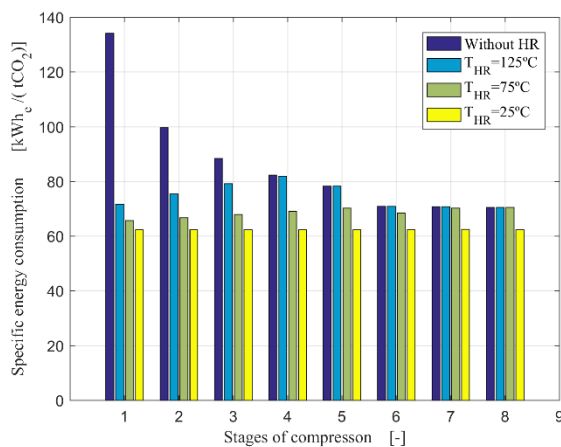


**Figure 10.** Waste heat from the CO<sub>2</sub> compression process at different temperatures.

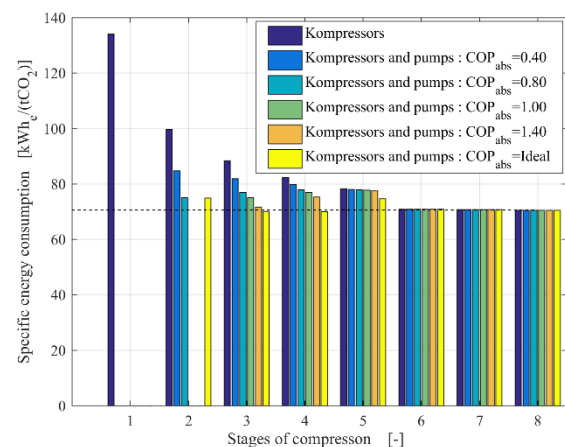


**Figure 11.** Maximum work which can be recovered by an ideal process from the waste heat produced in various CO<sub>2</sub> compression processes.

**3.1.3. Shockwave compression and waste heat recovery (refrigeration).** Figure 13 shows that this alternative is not more energy efficient than the *conventional* method, even when an ideal, reversible heat pump process is used as the basis for performance estimates.



**Figure 12.** Specific energy consumption of the compression process with Carnot heat recovery above three different  $T_{HR}$  temperatures.



**Figure 13.** Specific energy consumption using absorption refrigeration and  $n$  compressors with pressure ratios  $r = 150^{1/n}$ . Heat from CO<sub>2</sub> above  $T_{HR} = 100$  °C is used.

### 3.2. Engineering design study

The following heading present the results of the engineering design study work.

**3.2.1. Conventional compression and pumping cases.** Table 1 presents the results of the optimization study for the *conventional* compression cases.



**Table 1.** Optimisation parameters for *conventional* compression cases.

	Compressor pressure ratio, $r$			
	C1	C2*	C3	C4
<b>Stage 1</b>	2.00	3.98	3.78	3.78
<b>Stage 2</b>	2.00	2.00	2.45	2.46
<b>Stage 3</b>	2.00	2.00	2.00	2.01
<b>Stage 4</b>	2.00	1.92	1.83	1.83
<b>Stage 5</b>	2.00	1.72	1.66	1.66
<b>Stage 6</b>	2.00	1.52	1.45	1.45
<b>Stage 7</b>	1.86	1.24	1.21	1.19
<b>Specific power, <math>W_o</math> (kWh<sub>e</sub>/tCO<sub>2</sub>)</b>	91.9	89.2	89.0	89.0

\*Case C2 is taken to be the optimum case for the *conventional* compression process given the negligible performance gain achieved by adding one additional compressor wheel to this compressor design.

Table 2 presents some important performance results for the refrigeration and pumping cases based on the C2 compressor stage ratios shown in table 1.

**Table 2.** Summary of performance results for refrigeration cases.

Performance parameters	C2-4	C2-5	C2-6	C2-7
<b>Number of stages before pumping (-)</b>	4	5	6	7
<b>Refrigerant temperature, <math>T_{\text{refrig}}</math> (°C)</b>	-20.6	-0.60	10.0	-
<b>Refrigeration compressor pressure ratio, <math>r</math> (-)</b>	5.74	2.49	1.69	-
<b>Refrigeration unit efficiency, COP (-)</b>	3.78	7.80	14.0	-
<b>Specific power, <math>W_o</math> (kWh<sub>e</sub>/tCO<sub>2</sub>)</b>	96.2	90.2	87.7	89.2

\*for the seven-stage compressor design, the CO<sub>2</sub> condensing temperature is above 25 °C

**3.2.2. Optimization of shockwave compression cases.** Three ORC working fluids and two cycle cases are considered as part of the optimization study for both two and three-stage compression cases. To simplify the presentation of the results, each of the cases is assigned a number which is summarized table 3.

**Table 3.** Summary of shockwave compression optimisation cases.

ORC fluid	two-stage compressor		three-stage compressor	
	without IHE	with IHE	without IHE	with IHE
<b>Cyclopentane</b>	S1-2	S4-2	S1-3	S4-3
<b>nPentane</b>	S2-2	S5-2	S2-3	S5-3
<b>nButane</b>	S3-2	S6-2	S3-3	S6-3

Table 4 presents the results from the optimization study conducted for the shockwave compression cases based on the case summary presented in table 3.

**3.2.3. Overall performance comparisons.** Figure 14 and Figure 15 present a breakdown of overall specific power contributions for optimum design cases identified in tables 2 and 3: C2, S3 and S9.

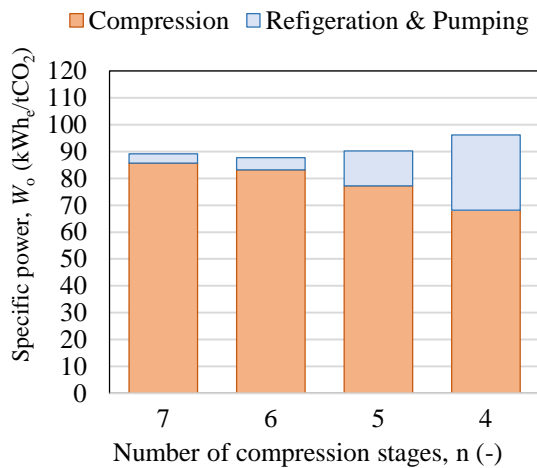
**Table 4.** Optimisation parameters for shockwave compression cases.

Performance parameters	S1-2	S2-2	S3-2 <sup>a</sup>	S4-2	S5-2	S6-2	S1-3	S2-3	S3-3 <sup>a</sup>	S4-3	S5-3	S6-3 <sup>b</sup>
<b>CO<sub>2</sub> compressor Stage 1, <math>r</math> (-)</b>	10.0	10.0	10.0	10.0	10.0	10.0	10.0	9.20	6.60	9.59	10.0	-

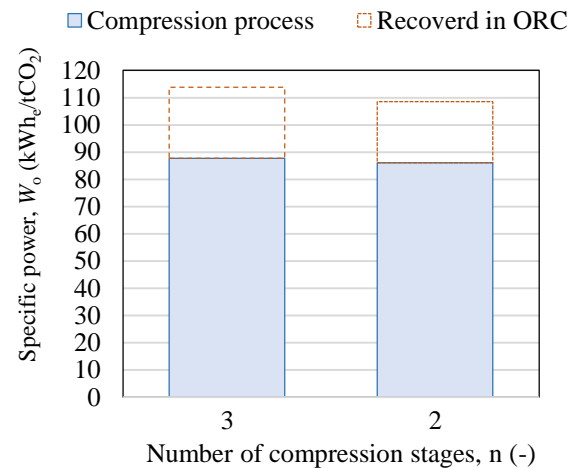
<b>CO<sub>2</sub> compressor Stage 2, <math>r</math> (-)</b>	8.69	8.69	8.69	8.69	8.69	8.69	3.40	4.80	5.00	3.23	4.13	-
<b>CO<sub>2</sub> compressor Stage 3, <math>r</math> (-)</b>	-	-	-	-	-	-	3.40	4.61	4.90	5.07	3.79	-
<b>ORC pressure, <math>P_{\max}</math> (MPa)</b>	1.20	3.04	3.53	1.33	3.04	3.53	1.83	3.04	3.53	2.10	2.60	-
<b>ORC temperature, <math>T_{\max}</math> (°C)</b>	149	188	152	154	188	145	174	189	147	186	178	-
<b>ORC condenser inlet, <math>T_{\text{cond}}</math> (°C)</b>	83.0	90.7	61.0	39.3	35.2	32.0	82.6	90.7	48.3	39.3	35.2	-
<b>ORC thermal efficiency, <math>\eta_t</math> (-)</b>	0.17	0.19	0.18	0.18	0.21	0.17	0.19	0.19	0.17	0.21	0.21	-
<b>Specific power, <math>W_o</math> (kWh<sub>e</sub>/tCO<sub>2</sub>)</b>	93.0	87.2	86.1	97.4	92.4	88.5	98.6	94.0	87.7	101	95.3	-

<sup>a</sup>Cases S3 & S9 represents the optimum case for the shockwave compressor

<sup>b</sup>In case S9 the condenser inlet temperature is close to the cooling temperature (48,3 °C cf. 25 °C) and therefore, the IHE performs no useful function and the results for case S12 are omitted.



**Figure 14.** Breakdown of overall specific power contributions for the C2-4, 5, 6 and 7 design cases.



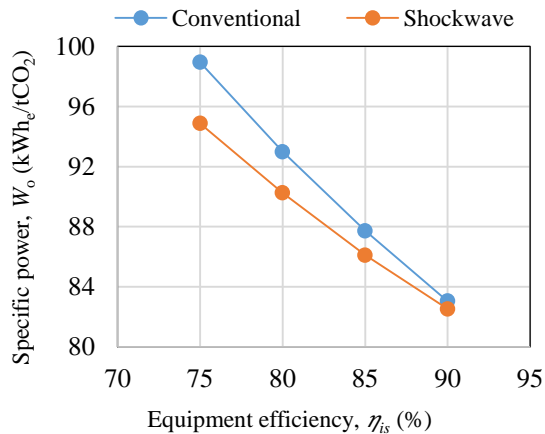
**Figure 15.** Breakdown of overall specific power contributions for the S3-2 and S3-3 design cases.

**3.2.4. Sensitivity study.** Figure 16-Figure 18 present variation in overall specific power with three key design parameters for the two best performing *conventional* and shockwave compression cases: C2-6 and S3-2.

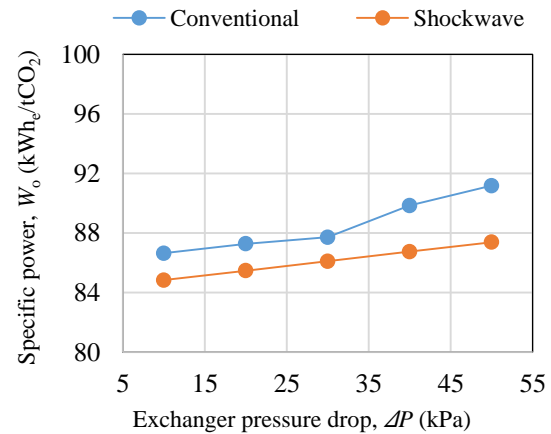
#### 4. Discussion

Figure 8 shows that for a *conventional* compressor design with constant stage pressure ratio the power consumption is within 10% of the theoretical minimum achievable power once the compressor has more than eight stages. Figure 9 shows that in the most optimistic case where CO<sub>2</sub> is condensed below critical pressure and pumped to the final pressure a maximum 10% saving in specific compression power is achievable, or that both cases have the same power given the most optimistic imaginable operating parameters.

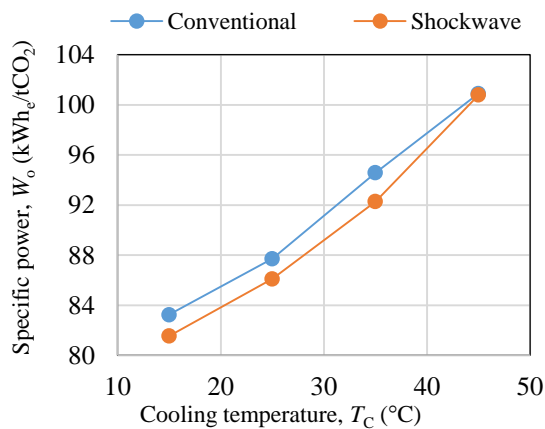
Figure 9 shows that the minimum power for a scheme where refrigeration and pumping is employed should occur after around five stages of compression and provide a small overall saving in power compared to a *conventional* eight-stage design. The results presented in Figure 14 for the optimized support this, but also shown that the saving in overall power for an optimized scheme is almost negligible.



**Figure 16.** Variation in overall specific power for the *conventional* (C2), and shockwave (S3) design cases with varying equipment efficiency.



**Figure 17.** Variation in overall specific power for the *conventional* (C2), and shockwave (S3) design cases with varying exchanger pressure drop.



**Figure 18.** Variation in overall specific power for the *conventional* (C2), and shockwave (S3) design cases with varying cooling temperature.

The savings in overall power achievable in a scheme employing refrigeration and pumping are dependent on the efficiency of the refrigeration unit. In the extreme case, where refrigeration duty is freely available, e.g. at a location adjacent to an LNG import terminal, there is an obvious benefit for this type of a design. In the present study a single-stage ammonia refrigeration unit has been assumed. Table 2 shows that the COP for this refrigeration unit varies between 3.78 and 14.0, with the compressor pressure ratio varying from 5.7 to 1.7. Although a pressure ratio of 5.7 in case C2-4 potentially indicates that a two-stage refrigeration process may offer lower overall power, Figure 14 shows that the case C2-6 provides lower overall power than cases C2-5 and C2-4, making further optimization of the C2-6 case redundant.

Figure 10 shows that only compressor designs with 3 or fewer stages generate a significant amount of waste heat above  $100^{\circ}\text{C}$ . Figure 11 and Figure 12 show that, although heat below  $100^{\circ}\text{C}$  can only be converted into work with low efficiency there is potential to recover some useful work from compression processes with up to 8 stages. This supports the findings of Farajollahi et al [9] who found that 3  $\text{MW}_e$  could be recovered from a six-stage compressor with a duty of 13.51  $\text{MW}_e$ . However, Figure 12 also shows that the overall specific power consumption for a compression process with heat integration is a minimum where the number of compression stages is also minimized and therefore that the energy saving reported by Farajollahi et al. [9] may be less significant if their design was compared to other, optimized, design alternatives.

The minimum practical number of stages achievable with a shockwave type compressor design is two. The optimization work carried out in this study for two and three-stage compression processes shows that pumping from above the critical pressure of CO<sub>2</sub> is still favorable for these cases. Figure 15 also shows that the optimum case is a two-stage compressor where the temperature level of the waste heat is highest and most beneficial use can be made in an ORC.

If the heat available in the shockwave compression cases can be used at a higher efficiency than that achieved in the present study the comparison of the *conventional* and shockwave results could be affected. The work of Bolland et al. [21] suggests that the use of this heat to replace steam extracted for boiler feed water heating at 200°C in a natural gas power plant could have a thermal efficiency up to,  $\eta_t = 0.3$  and that below 80°C the efficiency will drop to less than  $\eta_t = 0.1$ . Since the average thermal efficiency shown from Table 4 for the optimized ORC schemes developed here is  $\eta_t = 0.2$  and the average  $T_{\max} = 168$  °C, the potential gain from steam cycle integration can only be minor as would be the impact on the findings of this study.

The results of the sensitivity study presented in Figure 16-Figure 18 do not show any bias towards one of the two design cases considered here. They do, however illustrate the importance of compressor efficiency. If both compressors have a 90% efficiency the overall specific power consumption is almost equal and, over the range of efficiencies considered, the difference in efficiency separating the two compressors for any given overall power consumptions is only 2.5%. If the uncertainty in compressor efficiency is any greater than this, neither of the two cases has a clear advantage.

## 5. Conclusions

The results presented in this paper show that given equal compressor efficiency for both high pressure-ratio shockwave compressors and *conventional*, low pressure-ratio compressors, the minimum achievable power is essentially the same if both schemes are carefully optimized. The main conclusion of this study is, therefore, that the reduction in power consumption often claimed for CO<sub>2</sub> compression processes where either energy is recovered from waste heat in the compressor aftercoolers, or where CO<sub>2</sub> is condensed against a refrigerant and pumped to pipeline pressure from below its critical pressure is often optimistic. However, the result of the sensitivity study show that if one of these two alternatives have an advantage in terms of compressor efficiency, it is likely that this will translate into an advantage in terms of overall compression power requirements. Likewise, if a scheme using refrigeration and pumping has access to an external source of cooling, or if a shockwave compression process has a particular opportunity to use waste heat efficiently, this will play an important role in determining which process will be most efficient.

## 6. Highlights

- Comparative study of conventional and shockwave CO<sub>2</sub> compression
- Clear and unbiased comparison based on careful optimization of each case
- Concise presentation of results including assessment of the sensitivity to design parameters

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