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An Investigation of Turbomachinery Concepts for an Isothermal Compressor Used in an S-CO_2 Bottoming Cycle

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ABSTRACT

Various technology options are under progress to investigate the benefits of using an isothermal compressor for the s-CO_2 bottoming cycle applications. In previous works, the partial heating cycle has been investigated to show that there can be great increase in total net work when an isothermal compressor is applied to the system. The research covers the investigation on mainly three different turbomachinery concepts to realize the isothermal compressor, the radial-type compressor with impeller cooling, the multistage compressor with intercoolers, and the axial-type compressor with rotor and stator cooling. As a result, the concepts are mainly limited by the realistic cooling flux level that can be applied to the heat transfer surface, but the multistage compressor with intercoolers may be a viable candidate as long as the pressure drop in intercoolers remains low.

INTRODUCTION

To combat the issue of global warming, the development of various technologies is currently under progress to improve the performance of the conventional waste heat recovery systems. Various sources of waste heat, from glass manufacturing, steel manufacturing, and gas turbine exhaust, can be recovered to generate electricity by adopting another power cycle [1]. Among the candidates to enhance the current performance limits, the supercritical CO_2 power cycle (s-CO_2 cycle) has been considered a viable option to replace the conventional steam Rankine cycle, the system most often utilized for the bottoming cycle. Previous research on this technology has pointed out the potential to outperform the steam Rankine and organic Rankine cycles [2]. The reason that the s-CO_2 cycle achieves the advantage of high efficiency is that the system requires lower compression work due to compressing near the critical point, a region of high density and low compressibility [3].

In previous research efforts, the authors have devised a framework under which the performance of isothermal compressors in s-CO_2 cycle environment can be thermodynamically analyzed [4]. This turbomachine, although not yet fully developed for power cycle applications, is known to minimized compression work by reducing the temperature of the working fluid to isothermal conditions at the outlet. It has been proven that isothermal compressors can be beneficial for the case of s-CO_2 as a working fluid as they allow compression near the critical point. The reason that the isothermal compressor can be further useful as a bottoming cycle is that the isothermal compressor not only reduces the compression work but also lowers the compressor outlet temperature. Because the exhaust gas heat source from a topping cycle, mostly with high turbine inlet temperatures, is introduced at a given temperature and not the total heat, such a change in the cycle can still bring maximum net work.

In Cho et al. [5], a comparison of various s-CO_2 bottoming cycle layouts is presented as candidates for the bottoming cycle applied for the large-sized (100MW level) gas turbine heat source. In addition, research conducted by Kim et al. covers other s-CO_2 bottoming cycle layouts to compare the total net work produced [6]. One of them is the partial heating cycle, shown in Fig. 1, which can generate relatively high net work

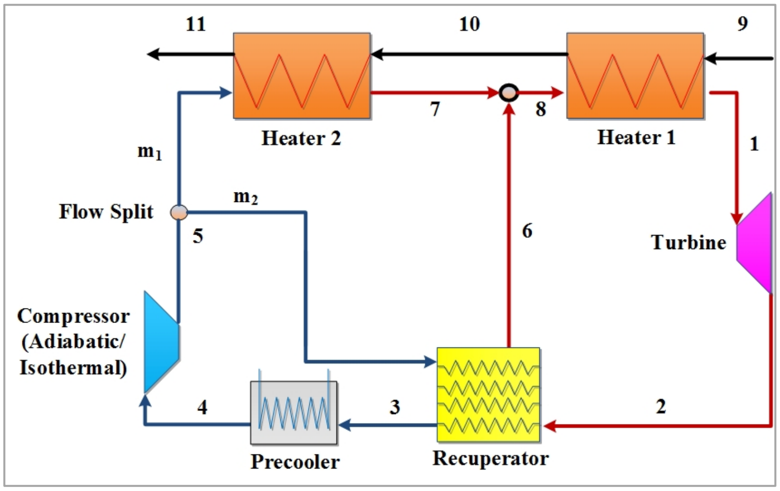


Fig. 1. Configuration of partial heating cycle

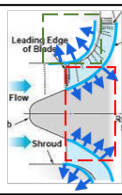
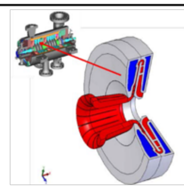
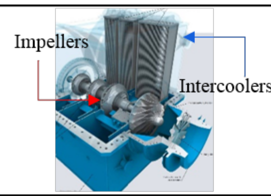
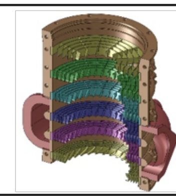
Type	Radial		Multistage + intercooling	Axial
Cooling method	Impeller and shroud surface cooling	Internally-cooled diaphragm [7]	Series of intercoolers	Rotor and stator blade cooling
Diagram				
Comments	- Cooled flow passes through the impeller and shroud surfaces to remove heat - Challenging cooling heat flux levels due to limited heat transfer area - Close to real isothermal compression	- Novel concept - Developed by Southwest Research Institute and Dresser-Rand - Removes heat of compression between each impeller	- Adiabatic compression followed by intercoolers - Realistic option for realizing s-CO_2 isothermal compressor (e.g. commercialized by MAN Turbo) - Large pressure drop expected in between intercoolers - Cycle re-optimization needed	- Cooling done on the rotor and stator blade surfaces - Concept from Frontline Aerospace IsoCool^{TM} - Larger heat transfer area at each stage - Axial compressor not yet realized for s-CO_2

Table 1. Various options of concepts to realize the isothermal compressor in s-CO_2 cycles

with a simple layout as well as a small number of components. The benefits of introducing the isothermal compressor to the partial heating cycle under the bottoming cycle conditions have been stated in [4], which mentions nearly 15% increase in net work through the replacement.

This research covers the investigation on what turbomachinery concepts would be best fitting to realize such an isothermal compressor, of which its importance and advantages have been stated previously. It becomes important not only to evaluate the performance using thermodynamic framework, but also to assess the isothermal compressor in the perspective of concept realization and physical feasibility. Out of many concepts for realizing the isothermal compressor, shown in Table 1, four main options have been explored. Firstly, in order to realize the concept on a radial-type machinery, which is the most designed type for the s-CO_2 cycle application, the method of impeller surface cooling can be considered as a novel approach to apply cooling to the s-CO_2 flow path. Although it could be challenging to apply high level of cooling flux on a small heat transfer area, it would still achieve close to real isothermal compression, if fully realized. Another novel radial-type concept, developed in collaboration between Southwest Research Institute and Dresser-Rand, would be to apply internal cooling on the diaphragm between impellers to enable a nearly isothermal compression [7].

Other concepts include the multistage compressor with intercoolers and axial compressor with rotor and stator blade cooling. Each concept has been deemed more realistic as similar concepts have been researched previously, multistage compressor with intercoolers by MAN Turbo [8] and axial-type compressor with stator vane cooling by Frontline Aerospace IsoCool™ [9]. The first realistic option has been commercialized for high pressure ratio applications such as CCS and ASU by companies such as MAN Turbo. However, it would not achieve high enough efficiency value in order to have significant gain over the pressure drop occurring due to the increasing number of intercoolers. Nonetheless, this option is selected for further investigation, to offer the most realistic guidelines for implementing the s-CO_2 cycle using isothermal compressors. Furthermore, another plausible option is to provide cooling for the rotor and stator blades in the axial-type compressor. This concept borrows the idea from the earlier mentioned Frontline Aerospace IsoCool™ [9] idea, but the possibility still remains questionable since the axial-type compressor has not been realized for s-CO_2 cycle applications, only by adopting a theoretical design software by Wang et al. [10]. This concept, too, would be investigated in key aspects of the component design which includes checking whether the level of cooling flux becomes reasonable.

In short, the research covers three main concepts for investigation at the conceptual level: the radial-type compressor with impeller surface cooling, the multistage compressor with intercoolers, and the axial-type compressor with rotor and stator blade cooling. The research aims to provide a general methodology and a set of guidelines in the design of s-CO_2 isothermal compressors in s-CO_2 bottoming cycle applications. Further details on how to implement the three concepts will be specified in the following sections.

OPTION 1: RADIAL-TYPE COMPRESSOR

Firstly, the idea of radial-type impeller surface cooling can be compared to the previous research on impeller cooling conducted by Moosania et al. [11]. It was concluded that the most effective cooling place to increase the performance and to reduce the impeller pressure is the compressor shroud surface. In a similar manner, this research focuses specifically on investigating a novel concept of realizing the isothermal compression through cooling the impeller hub and shroud surfaces of the radial-type compressor. In order to assess its feasibility, a simplified approach is firstly adopted to estimate the required cooling flux on the compressor surfaces. If the magnitude of its cooling flux reaches too high, then it would become physically unfeasible to design an isothermal compressor concept. The cooling flux profile would also be essential for running the computational fluid dynamics (CFD) analysis of this concept, to observe the change in the internal flow behavior.

In order to obtain information for evaluation, several assumptions regarding the geometry and the heat removal need to be made. Both the infinitesimal heat profile using the infinitesimal approach from previous works [6] and the infinitesimal area profile from given geometry are needed to obtain the cooling flux level, $q''(r) = \frac{q(r)}{A(r)}$. This study takes the geometry information from the s-CO_2 compressor of the SCIEL loop facility in KAERI [12] as a reference. The rationale behind this is to utilize the existing information on compressor geometry so that a more realistic analysis can be performed. Details of the SCIEL design conditions are shown in Table 2. Then, the area profile assumes that the diameter curve from the inlet to the outlet is 2nd order polynomial, and the area profile and the infinitesimal heat profile are matched as a one-to-one relationship.

The results of the radial-type compressor with impeller cooling concept are evaluated. Fig. 2 displays the result of the calculation performed to obtain the required cooling flux information. However, as shown, the flux level reaches to nearly several hundreds of MW/m^2, which is a level physically unfeasible even when compared with the level achieved by the state-of-the-art technology for cooling flux [13]. First observation made from this analysis is that the required cooling flux levels peak rapidly at the entrance, especially since

Design parameters	Values
Mass flow rate (kg/s)	1.1
RPM	70000
Inlet stagnation temperature (°C)	33
Inlet stagnation pressure (MPa)	7.8
Pressure ratio	1.8
Inlet diameter	23mm
Outlet diameter	46mm
Length	22mm
Number of blades	16
Isentropic efficiency (%)	65
Compression process number	100

Table 2. Design variables obtained from SCIEL conditions and isothermal compressor methodology [12]

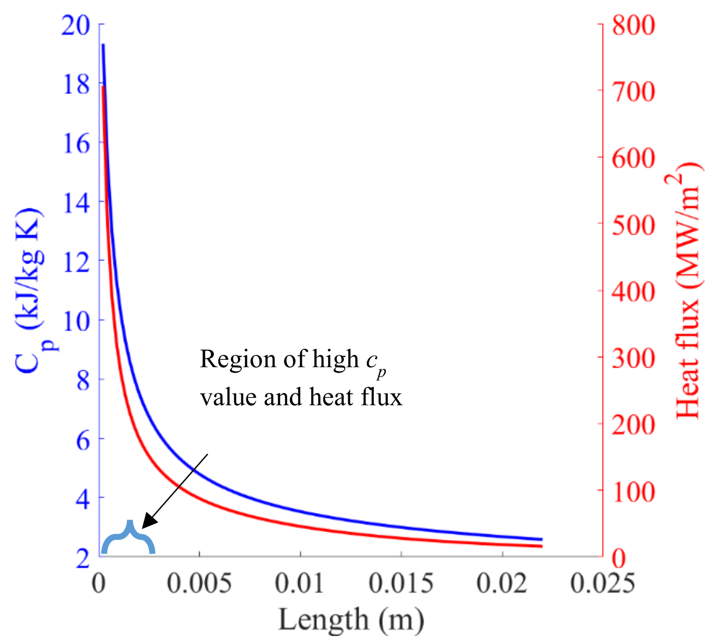


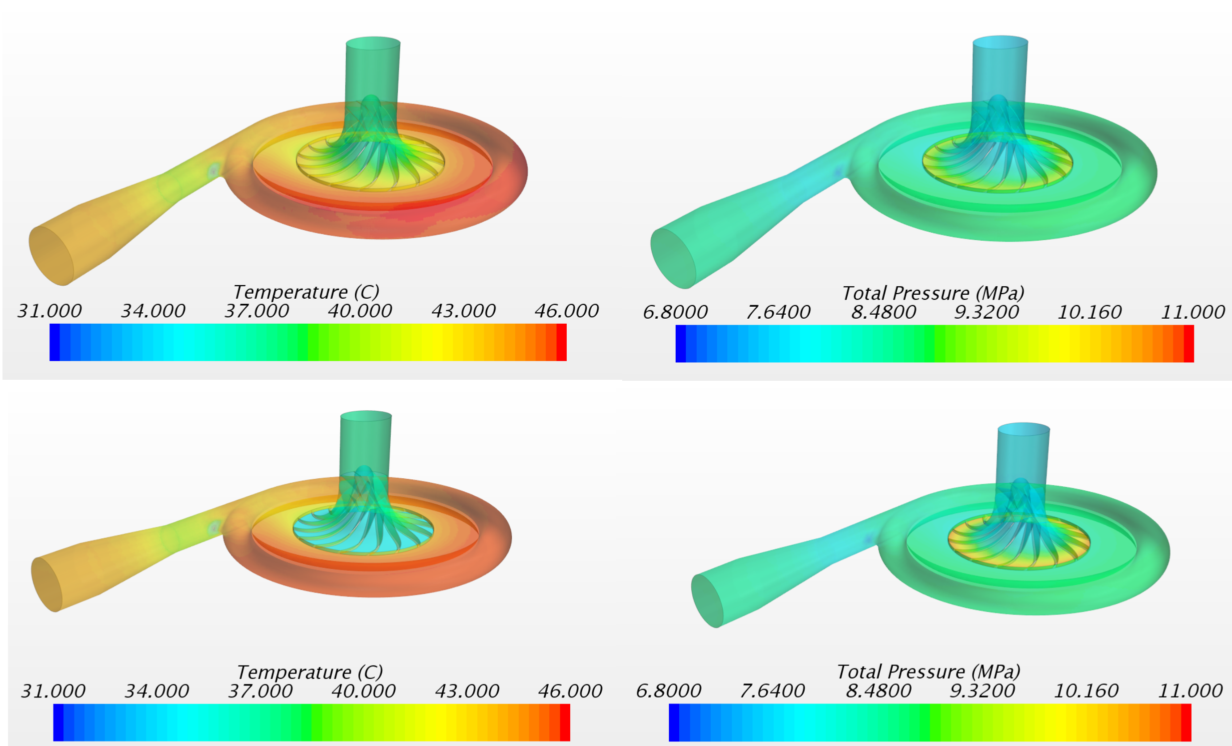
Fig. 2. Calculated specific heat (c_p) and heat flux profile within the radial-type isothermal compressor

the radial compressor has a low inlet flow area that demands more heat removal at higher flux level within the given surface dimensions. Second comment is that because the inlet condition is near the critical point, specific heat of the working fluid climaxes exponentially specifically at the inlet and decays to much lower levels towards the outlet.

To progress further in the investigation, a simple computational fluid dynamics (CFD) analysis is conducted using a commercial code STAR-CCM+ V11.06. The simulations have adopted the $k-\omega$ SST (Shear Stress Transport) model with wall function, and a CSV table of s-CO_2 thermodynamic properties has been set up to read off from NIST REFPROP database [14]. The mesh scheme consists of polyhedral elements with 10 prism layers. Furthermore, the total inlet temperature has been raised to 38°C in order to achieve better convergence for the results. To investigate the changes in compressor performance, 2 different sets of thermal specification conditions are provided to the reference compressor. The combination of thermal boundary conditions is the following: adiabatic (case 1), and constant temperature for hub and shroud surfaces at 35°C (case 2). The comparison of these two cases is analyzed for temperature and total pressure, and it is displayed in Figs. 3-6 below.

The results show that even if a reasonable thermal boundary condition is provided to the compressor, the pressure and temperature fields would only change locally and fail to yield the expected outlet conditions. It can be seen that by cooling the hub and surfaces, the total pressure increases near the impeller tip, up to nearly 10.5MPa, compared to 9.5MPa for the adiabatic case. The discharge temperature for the adiabatic condition is 47.9°C, and that of the constant temperature condition is 46.3°C. It can also be said that the overall temperature after the impeller exit is not cooled. This is evidently due to the insufficient cooling provided by the thermal boundary condition on the surfaces. Hence, providing the right thermal boundary conditions is essential in simulating the isothermal compression for the CFD analysis.

In short, the radial-type compressor may not be favorable in the design perspective of the cooling flux required for enabling isothermal compression. Furthermore, because the radial-type compressor is designed for lower capacity systems, the surface area cannot be increased greatly due to the expected pressure drop increased at the flow path by additional fin structures. Thus, radial-type compressors may not offer sufficient surface area for cooling required for isothermal compression.



Figs. 3-6. Temperature and total pressure of SCIEL s-CO_2 compressor under adiabatic conditions (top left and right) and constant temperature condition at hub and shroud surfaces (bottom left and right)

OPTION 2: MULTISTAGE COMPRESSION WITH INTERCOOLING

Secondly, the multistage compressor with intercooling concept can be evaluated of how closely it can be modelled as an isothermal compressor, although not perfectly isothermal. Fig. 7 represents an example of what a 4-stage compression with intercoolers would look like. The main limitations of efficiency are the pressure drop in intercoolers and the restrained number of compressor stage number due to the overall component size and cost. Hence, the performance analysis of this concept would be to assess the isothermal compressor efficiency, η_{iso-c}, with respect to the stage number and intercooler pressure drop.

Fig. 8 shows the results of analyzing the concept of multistage compression with intercooling. The intercooler exit temperature has been assumed to be equal to the compressor inlet temperature. Furthermore, the pressure ratio has been divided so that each compressor is allocated with $(PR)^{1/stage\ number}$. The T-s diagrams of the cases for stage number 5 and 10 reveal how the outlet temperatures vary up to 2°C due to the differences in the intermediate pressure ratios of compressor stages. Furthermore, the performance indicator η_{iso-c} increases with respect to the compressor stage number when dP = 0kPa (ideal case), but as dP takes a higher value, the efficiency peaks at a lower stage number. It can be seen that if the pressure drop can be minimized in the intercoolers, the multistage compression with intercooling concept can be a strong candidate for achieving near-isothermal compression.

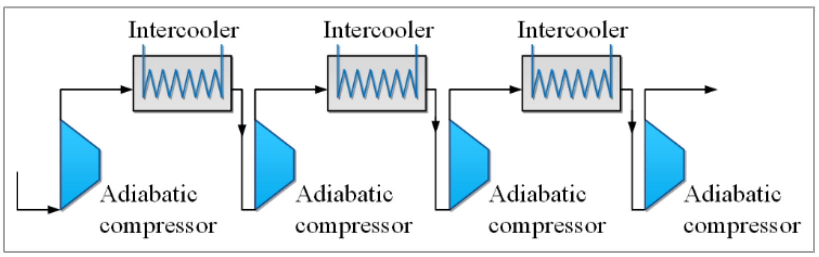


Fig. 7. Multistage compression with intercooling (4 stages)

Design Parameters	**Values**
Compressor inlet total temperature (°C)	32
Compressor inlet total pressure (MPa)	7.69
Total-to-total pressure ratio	2.6
RPM	3600
Compressor mass flow rate (kg/s)	1915
Realizable cooling flux level (MW/m^2)	5
Compression process number	50
Isentropic efficiency (small stage efficiency) (%)	89

Table 3. Design parameters involved in the analysis of axial-type compressor [10]

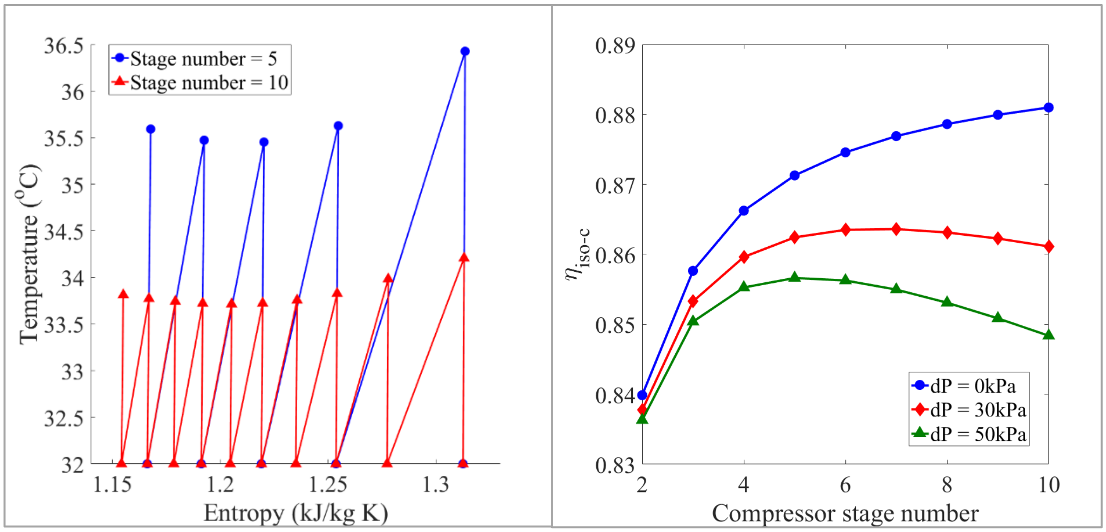


Fig. 8. T-s diagram (left) and isothermal compressor efficiency (right) for multistage compression with intercooling

OPTION 3: AXIAL-TYPE COMPRESSOR

Lastly, in order to assess the overall design and the physical feasibility of the axial-type isothermal compressor, the design data from Wang et al. [10] is adopted for its geometric considerations. Using these reference values, shown in Table 3, the surface area of the rotor and the stator at each stage is then summed up to create a profile. The information is required to apply the maximum cooling flux level that ultimately yields the total cooling capacity of the axial-type isothermal compressor, given the amount of heat removal required for perfect isothermal compression. This also assumes that the method of cooling would be introduced at the blades, on which the means of heat transfer pathways are embedded. The concept, although not yet realized, can be derived from a similar gas turbine cooling technology developed by Frontline Aerospace named IsoCool™ [9].

In order to implement this analysis, it adopts a major assumption coming from the realistic cooling flux level, referring to a high-flux cooling technology using supercritical fluids including s-CO_2 [13]. Referring from this research, it can be stated that a physically realizable cooling flux level can be taken as 100-500W/cm^2 (or 1-5MW/m^2). Hence, this value can be used together with the reference geometry to produce the results of the realistic isothermal compression, displayed in Fig. 9. It explains the pathway for a realistic isothermal compression process incorporating the cooling flux constraint, shown by the reduced cooling process arrows marked in blue. Numerically, the compression work would be less than that of the adiabatic compression process marked in $1 \rightarrow 2'$, but is expected to be higher than that of the perfect isothermal compression process marked in $1 \rightarrow 2$.

The analysis results for the axial-type compressor concept show that given the total surface area available by the rotor and stator blades, the total heat removal cannot suffice the amount for perfect isothermal compression. When taking the reference geometry and the assumed cooling flux level as 5MW/m^2, and dividing the total amount of heat by the mass flow rate, the calculations yield 6.4kJ/kg of heat, while the perfect isothermal compression requires 64.8kJ/kg of heat removal. Under the conditions of Table 3, the values of isothermal compressor efficiency, η_{iso-c}, result in 75.9% for the realistic isothermal compression, in comparison to 74.7% for the adiabatic compression and 88.9% for the perfect isothermal compression. This converts to nearly 0.3kJ/kg specific work reduction in compressor work for the realistic isothermal

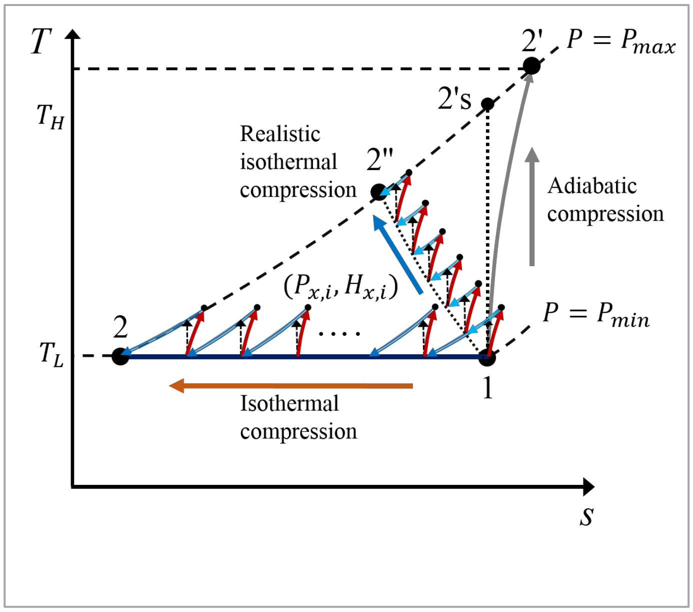


Fig. 9. T-s diagram explaining the thermodynamic pathway of realistic isothermal compression using the infinitesimal approach

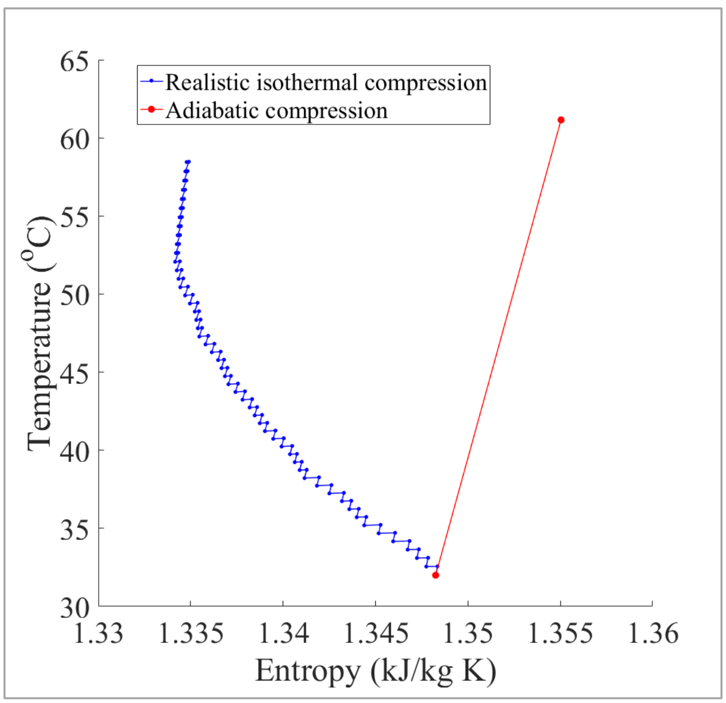


Fig. 10. T-s diagram comparing the realistic isothermal compression to the adiabatic compression under the given design conditions of Table 1

compression, whereas the perfect isothermal compression reduces the specific compression work by 3.2kJ/kg. Therefore, it can be seen that the limitation in cooling flux level can severely hinder the potential for reducing compression work for the s-CO_2 isothermal compressors. Fig. 10 plots the thermodynamic pathway of the realistic isothermal compression on the T-s diagram in comparison to the adiabatic compression.

SUMMARY AND CONCLUSIONS

In conclusion, three main concepts have been evaluated for the possibility of implementing the isothermal compressor for s-CO_2 bottoming cycle applications. The radial-type compressor with impeller and shroud surface cooling, along with the axial-type compressor with rotor and stator blade cooling, becomes restricted mainly by the cooling methods to achieve isothermal compression requiring significant heat removal. Because of the restraint on cooling flux due to the limited surface area for heat removal, the concepts either become impossible to implement or have to resort to a realistic isothermal compression, at the cost of reduced performance. On the other hand, the conventional concept of multistage compressor with intercoolers has also been evaluated under the methodology suggested, and it can be modified to suit best the conditions for s-CO_2 cycle. In short, the isothermal compressors can be brought closer to realistic design if the thermal boundary constraint of cooling flux level can be applied at a reasonable level.

Further works include the investigation of cooling flux level appropriate for the isothermal compressor concepts. This may include the heat exchanger design integrated to the turbomachine geometry and the use of novel technologies for turbomachinery design to enable a near-isothermal compression process.

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