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Radial Compressor Design and Off-Design for Trans-critical CO_2 Operating Conditions

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ABSTRACT

Supercritical carbon dioxide (sCO_2) cycles provide good potential for power generation with high efficiency and low emissions. The highest cycle design efficiency and power output typically results when the main compressor for these cycles has inlet conditions near the critical point. These inlet conditions pose a challenge in the design and off-design analyses as the fluid is typically far from satisfying ideal gas and calorically perfect gas conditions typical of most compressor design and off-design procedures. In this study, a multi-staged intercooled centrifugal compressor design is made for an indirect-fired sCO_2 cycle, and the design point data are used to generate the performance maps using a 2D mean-line analysis code. Fluid and turbomachinery similarity laws are applied to find an appropriate surrogate ideal gas fluid to be used in the design and performance map generation codes. The performance maps are then scaled back for the sCO_2 fluid case using the affinity laws. The developed methodology is validated using the experimental data for a compressor operating under similar inflow conditions.

NOMENCLATURE

c_p	Specific Heat at Constant Pressure
d_2	Impeller Diameter
e_c	Polytropic Efficiency
h	Enthalpy
H	Isentropic Head
P	Pressure
$\dot{m}$	Mass Flowrate
n_s	Specific Speed
Pr	Prandtl Number
R	Gas Constant
Re	Reynolds Number
T	Temperature
$\dot{V}$	Volumetric Flowrate
V	Speed
Z	Compressibility Factor
α	Thermal Diffusivity
ε	Slip Factor
ρ	Density
π_c	Compressor Pressure Ratio
γ	Ratio of Specific Heats
γ_{PV}	Isentropic Volume Coefficient
γ_{PT}	Isentropic Temperature Coefficient
μ	Dynamic Viscosity
ν	Kinematic Viscosity
η	Isentropic Efficiency

Subscript t denotes total properties

INTRODUCTION

The objective of this paper is to present a methodology developed for the design and off-design operations of compressors operating in the trans-critical carbon dioxide (CO_2) flow region. However, indicated trans-critical conditions also correspond to the point at which several of the thermodynamic quantities such as specific heat capacity, specific heat ratio, and compressibility factor change drastically, which makes use of traditional compressor design methods inapplicable for these inlet conditions. Several methods from the open literature for supercritical CO_2 (sCO_2) compressor design were analyzed and compared in this study. Due to the high gradients in the thermodynamic properties at the design inlet flow conditions, design methodologies such as real gas analysis involving the use of tables and interpolations needed to be phased out for this study. Fluid similarity methodology is used to perform the design and off-design analyses of a compressor operating under trans-critical CO_2 inlet conditions. The off-design maps generated with the similar fluid were then scaled to CO_2 conditions using turbomachinery affinity laws, and a compressor model was developed using these off-design maps for off-design sCO_2 cycle simulations.

The components and configuration of the 550 MW sCO_2 cycle used in this study are shown in Figure 1. This is an indirect-fired re-compression Brayton cycle with turbine reheat in which the heat source (not shown in Figure 1) is a coal-fired, oxygen blown circulating fluidized bed. It is modified from the classic re-compression Brayton cycle to enhance low-temperature heat recovery from the flue gas. A detailed description of the power cycle and optimization methodology can be found in a recent paper [1].

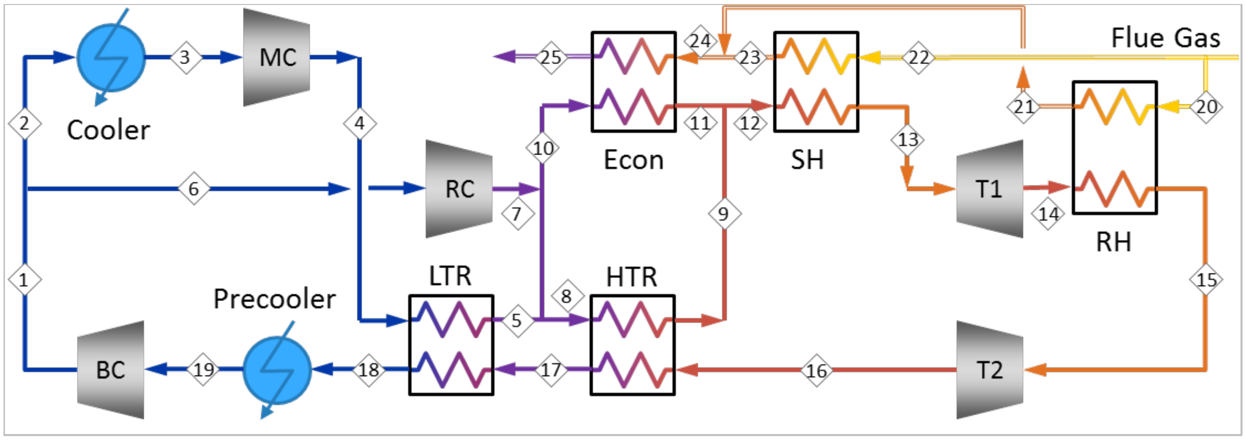


Figure 1. The configuration of the sCO_2 cycle depicting the trans-critical inlet flow compressor

The optimized state point corresponds to the minimum levelized cost of electricity. This state point yields trans-critical inlet flow conditions at the main compressor (MC). The cycle design calculations estimate the MC power to be within 99–130 MW. MC operating conditions and parameter ranges are provided in Table 1.

Table 1 Main compressor operating conditions

Parameter	Inlet	Exit
Pressure (psia)	887–1200	4096–5980
Temperature (F)	71–94	98–131
Mass Flowrate (lbm/s)	5326–6991	
Volumetric Flowrate (ft^3/s)	134–206	
Density (lbm/ft^3)	37–46	55–58

The ranges for MC operation in Table 1 are due to different cycle variations currently being studied at the National Energy Technology Laboratory (NETL) for the sCO_2 cycle, such as re-compression cycle with and without turbine re-heat, and partial cooling cycle with and without turbine re-heat.

In the optimized cycle system studies at NETL, the MC is considered to have two stages, where intercooling is applied between the stages and brings the second-stage inlet flow temperature back to the first-stage inlet temperature with about a 1% loss in pressure from the Stage 1 exit pressure.

The operation points of the MC were checked for the direction of the phase transition before beginning the design. Various compressor options for the studied indirect cycles and options gather around three different operation paths, as can be seen in Figure 2.

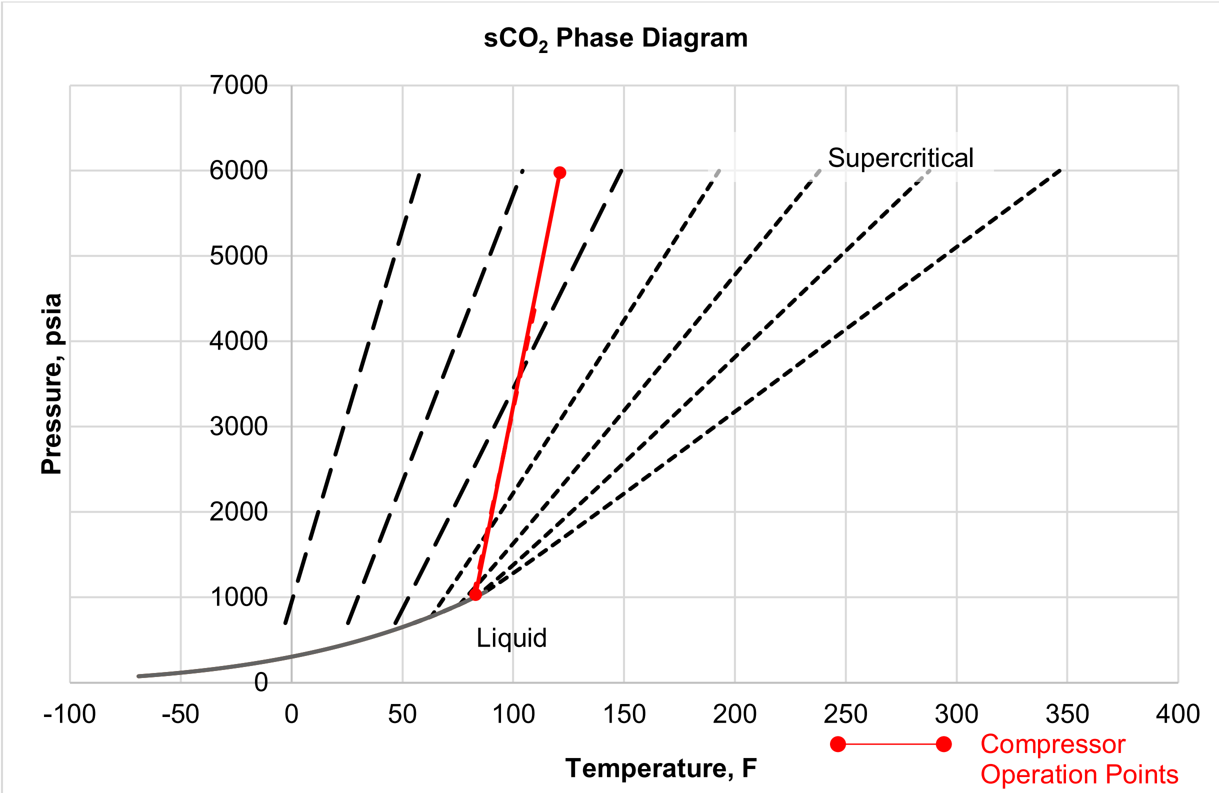


Figure 2. The direction of the phase transition for the considered compressor options

The sCO_2 phase diagram is used here to understand the direction of the phase transition in the compression and also check for the inlet conditions. The analysis shows that all compression options would operate from liquid to supercritical, where the phase change will occur inside the compressor. Note that while there is a phase change, there is no two-phase flow condition as in a transition from liquid to vapor, but rather, the fluid temperature and density increase and behave as a single phase. The National Institute of Standards and Technology (NIST) Reference Fluid Thermodynamic and Transport Properties Database 10 (REFPROP 10) [2] is used to calculate the compressibility factors to determine if the ideal gas assumption will hold at some portion of the compression. The results showed that ideal gas assumptions will not be valid for the entire compressor operation range.

The primary objective of this study is to generate off-design performance maps for the MC. Regardless of the performance map type, all maps need to be corrected for varying inlet state conditions; however, corrections typically assume a constant specific heat ratio [3, 4, 5]. To understand if calorically perfect gas assumptions will be valid for the MC design and off-design analyses, the change of specific heats of CO_2 flow from the inlet to the exit of each compressor option was analyzed using REFPROP 10 [2] (Figure 3).

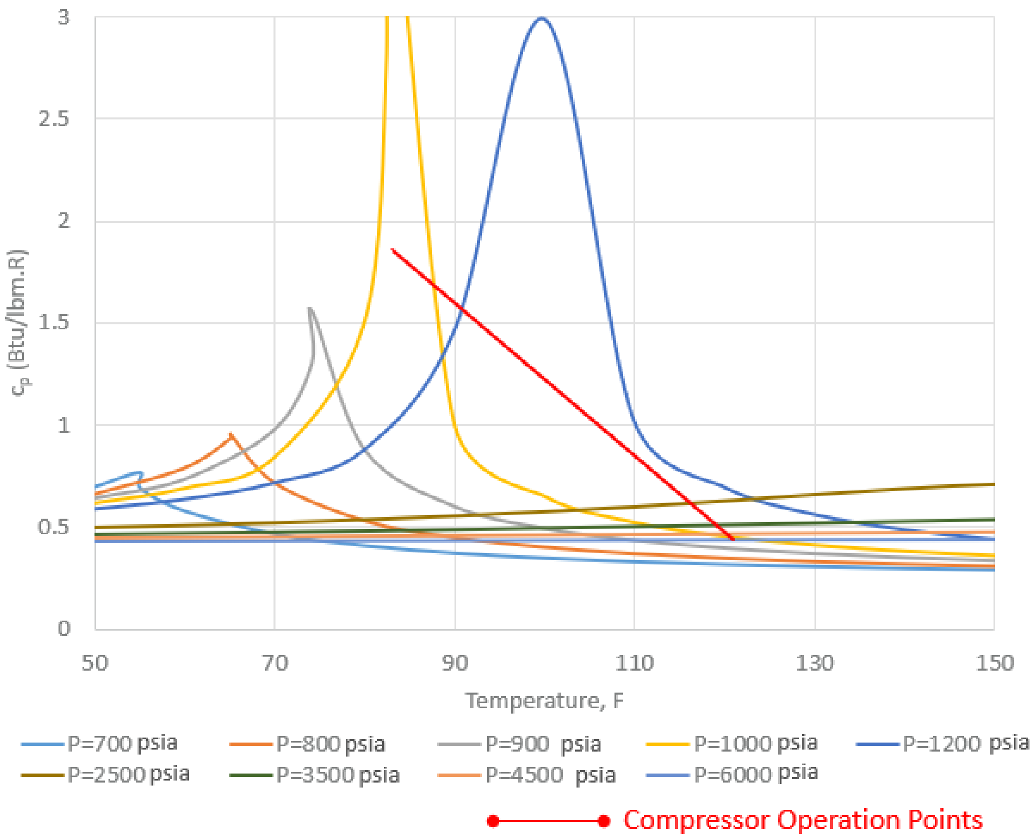


Figure 3. The change in the constant pressure specific heat coefficient from inlet to exit of each compressor option

For all compressor cases, the change of the specific heat (c_p) per compression stage was calculated to be 58–83% (absolute), which is well above the limits for calorically perfect gas assumption. An example application where this assumption holds is the air compressor cases, in which the rate of change of c_p per stage is less than 5%. Therefore, the methodology chosen to design and analyze off-design behavior of the MC should be one that can handle non-ideal gases with high gradients for the specific heats.

DETERMINING THE COMPRESSION METHOD

The exact type of the machine should be determined before starting the design analysis as it impacts the design methodology to be used, performance map generation methods, and off-design analysis [3].

Baije charts are typically used to initially determine the type of the compression or expansion type [5]. The type of the compression method is dependent on the mass (or volumetric) flowrate to the component, the inlet pressure and temperature, desired compression ratio, shaft speed, and the dominant phase of the incoming flow. The x-axis of the Baije chart is used to determine the compression type, which lists the specific speed (n_s) [6, 7].

The specific speed can be calculated by Equation (1).

$$n_s = \frac{\omega\sqrt{\dot{V}}}{(H)^{3/4}} \tag{1}$$

In Equation (1), ω is the shaft speed, $\dot{V}$ is the volumetric flowrate at the inlet, and H is the isentropic head coefficient, which can be calculated by Equation (2) [7].

$$H = \frac{\bar{z}RT_{t,in}}{\gamma - 1/_{\gamma}}\left[\pi_c^{\gamma-1/_{\gamma}} - 1\right] \tag{2}$$

In Equation (2), $\bar{z}$ is the average compressibility factor (between inlet and exit), R is the gas constant, $T_{t,in}$ is the total inlet flow temperature, γ is the ratio of specific heats (at the flow inlet (GPSA method) or as an average between inlet and exit (ASME method)), and π_c is the compression ratio. All inputs required for Equation (1) and Equation (2) to calculate the specific speed can be easily obtained from the cycle state point data and the desired shaft rotation speed. The state point data can be used in a thermodynamic property calculator to obtain other inflow thermodynamic properties required for the calculation. With the state point data presented in Table 1, the specific speed is calculated as presented in Table 2.

Table 2. Specific speed calculation

Stage	$\dot{V}$ (ft^3/s)	$\bar{z}$	$T_{t,in}$ (F)	π_c	γ	H (ft.lbf/lbm)	n_s
Stage 1	135	0.233	71.44	2.484	4.479	5744	1.057
Stage 2	114	0.488	71.44	2.506	2.464	11114	0.6

The calculated values for specific speed are then put on the x-axis of the Baije chart to obtain the compression options. As can be seen in Figure 4, these values correspond to two options for the compression method: centrifugal pump or radial compressor.

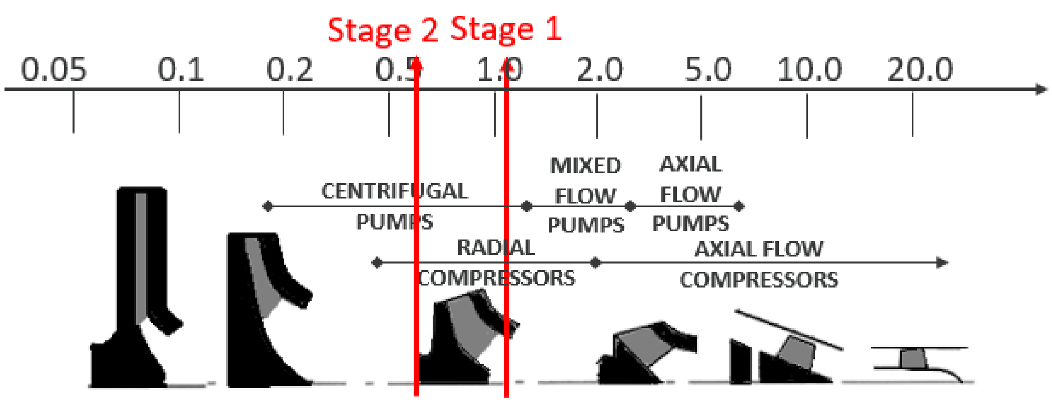


Figure 4 Determining compression type using specific speed

Between these options, pump should be preferred over compressor if the majority of the flow phase is in liquid phase. The analysis with the state points indicates that for both stages, inlet phase is liquid, but the exit flow phase is supercritical, and the phase change will occur at a point before the first-stage impellers. Therefore, radial compressor can be selected over pump because the majority of the flow phase is well above the critical point.

IMPLEMENTING REAL GAS EFFECTS INTO DESIGN AND OFF-DESIGN CALCULATIONS

Radial compressors are common in two-phase applications especially in the oil and gas industry—known as "wet-gas operation" [8]. Most of the design and off-design methods that use ideal gas assumptions could give unrealistic results if a real gas methodology is not used. Aungier [9] suggests three approaches to implement real gas effects into design and off-design analysis: a simple approach, pseudo-real gas approach, and full real gas approach. The differences between these approaches are summarized in Table 3.

Table 3. Real gas analysis methods applicable to compressor design and off-design analyses

Approach	Equation of State	Isentropic Flow Relations	Thermodynamic Properties	Computational & Formulation Complexity
Simple Approach	Ideal Gas	Ideal Gas	Real Gas-Tabulated	Simple
Pseudo-Real Gas Approach	Real Gas			Moderate
Full Real Gas Approach	Real Gas	Real Gas		Advanced

The pseudo-real gas approach, which retains the isentropic flow relations in the same form with ideal gas, uses real gas equation of state with compressibility factor and tabulated real gas properties. This approach is more popular than the other two because it has enough accuracy and is easier to apply than the full approach. Theoretically, the accuracy of the full real gas approach is expected to be better than the other two, but its application requires changing some formulas, re-derivation of some equations, and adding extra calculation steps for specific heat ratio definitions.

Conversion of ideal gas-based thermodynamic relations to real gas relations for the isentropic flow relations, such as total to static temperature and pressure ratios and critical area relations, is studied in detail by Nedersigt [10]. The biggest difference between ideal and real gas flow equations is the implementation of the ratio of specific heats (γ) and the compressibility factor (z).

In the case of ideal gas, the ratio of specific heats, or gamma, has a simple definition, which is the ratio of constant pressure to constant volume specific heat values. Whereas for the real gases, the ratio of specific heats has two different definitions as shown in Equation (3i) and Equation (3ii).

$$\gamma_{PV} = \gamma \left[\frac{z + T\left(\frac{\partial Z}{\partial T}\right)_v}{z + T\left(\frac{\partial Z}{\partial T}\right)_P} \right] \tag{3i}$$

$$\gamma_{PT} = \left[1 - \frac{R}{c_p}\left(z + T\left(\frac{\partial Z}{\partial T}\right)_P \right) \right]^{-1} \tag{3ii}$$

Each form of heat capacity ratios is used in different isentropic real-gas relations. For example, γ_{PV} is used for calculating the total to static pressure ratio (see Equation (4)), whereas γ_{PT} is used to calculate the temperature ratio from the pressure ratio (see Equation (5)).

$$P_t/P = \left(1 + \frac{\gamma_{PV} - 1}{2} M^2\right)^{\gamma_{PV}/(\gamma_{PV}-1)} \tag{4}$$

$$T_t/T = \left(P_t/P\right)^{\gamma_{PT}-1/\gamma_{PT}} \tag{5}$$

The temperature and pressure ratio relationships are frequently used in compressor design and off-design equations. When the design equations are to be replaced with the real gas equations, the value of the partial derivative $\left(\frac{\partial Z}{\partial T}\right)$ should either be calculated or obtained from a table. A simple REFPROP analysis to understand the rate of change of these partial derivatives at the inlet flow conditions of the MC was performed. The slopes of Z with respect to temperature are calculated from isoproperty tables generated for constant pressure and density, respectively. The results showed that the value of the constant pressure derivative $\left(\frac{\partial Z}{\partial T}\right)_P$ is highly variant, but the value of the constant density derivative $\left(\frac{\partial Z}{\partial T}\right)_v$ is almost constant. There is also a discontinuity in the constant pressure derivative, which would cause problems in iterations if its tabulated values are used.

Given the complexity of the full-real gas approach, it is important to know how much more accurate this particular method is. In Nedersigt's study on deriving real-gas isentropic flow relations [10], an estimation of the difference in several compressor exit state variables is provided for using the simple approach using REFPROP only to estimate them versus calculating them from the full real-gas relations. The calculations are done for Sandia National Laboratories' (SNL) sCO_2 compressor flow conditions [11], which are comparable to the MC conditions of this current study. The comparison showed that the greatest difference occurred in exit state density calculation by 4.38%, in exit total enthalpy by 1.85%, and exit static temperature by 0.28%. Use of the pseudo-real gas approach could reduce these differences from the full real gas calculations further. The full real gas approach will necessitate using tabulated thermodynamic properties, which would cause problems in data interpolations especially if the analyzed conditions are at the phase transition point. Therefore, using simpler property approaches are sometimes more favorable to the full approach to generate fewer errors if the thermodynamic properties have high gradients at the analysis conditions [11].

Various approaches for handling a non-isocaloric, real gas are present in the sCO_2 literature. SNL performed a compressor design for the experimental phases of their project. The compressor inlet state at design is supercritical, but off-design conditions might be trans-critical [11]. The design uses a surrogate fluid that is "similar" in terms of the volumetric flowrate, Reynolds number, and isentropic head coefficient. The surrogate fluid is an ideal gas and is used with the National Aeronautics and Space Administration's (NASA) Centrifugal Compressor Off-Design Program (CCODP) code [12] to produce performance maps. Then the fluid similarity laws are used again to scale the output maps to the sCO_2 flow. In the calculation process, REFPROP is used for gas properties. Pressure ratio and efficiency versus mass flowrate charts are obtained for different speeds and compared with experimental data, which show significant agreement for the efficiency curves (a ~5% difference).

In the study by Pelton et al. [13], the design inlet flow conditions are above the critical point; the compressor is, therefore, operating in the supercritical region. A 2D design is performed with averaged real gas properties between the inlet and exit of each compressor station, which yield a shrouded impeller design with the indicated flow coefficient value and impeller dimensions. For the design, the simple approach is used, whereas in the off-design analysis, pseudo-real gas approach is used. A Computational Fluid Dynamics (CFD) Solver with Spalart-Allmaras turbulence model is used in generating the off-design maps, where all inlet conditions studied are above critical point. Head and efficiency versus flow coefficient curves are plotted for three different inlet pressures.

A compressor performance analysis study by Liese et al. [4] focuses on a compressor in a 10 MW plant operating at the critical point. No intercooling is used in this two-stage compressor design, and the design calculations use exit flow coefficient with real gas isentropic flow relations. This study uses a full real-gas approach. The off-design analysis uses correlations for efficiency and head coefficient but with the compressor exit flow coefficient, instead of the inlet flow coefficient. Efficiency and isentropic head versus flow coefficient curves are generated.

In the He.Ro.eu project [14], where the sCO_2 cycle is used as a bottoming cycle for a nuclear power plant, a radial compressor design is made for the supercritical inlet conditions. The design method uses tabulated real gas properties with real gas equation of state; therefore, a form of that pseudo-real gas approach is used. Performance curves are then calculated with ANSYS-CFX CFD software with the Mentor SST k-omega turbulence model. Pressure ratio and efficiency versus volumetric flowrate curves are calculated for different shaft speeds.

An experimental study by Southwest Research Institute with two-phase inlet sCO_2 flow [8] has analysis conditions similar to the current study. The design inlet conditions include a 24.5% inlet liquid mass fraction. Their study is a preliminary analysis for a more detailed future experimental study that would involve a compressor test set up. An analytical black-box compressor model is used to understand the impact of using two-phase inlet flow on compressor performance. The results show that when the inlet flow is mostly liquid, the compressor efficiency drops to half of its design value for dry gas inlet. The performance curves will be calculated with experiments, which will be pressure ratio and efficiency versus two-phase inlet flow coefficient.

In summary, the literature review of different studies shows that most of the compressor design studies with real gas flow conditions either use a pseudo-real gas or full real gas approach. In all studies, property tables are used for real gas thermodynamic properties. It should be noted that except for SNL's study, the inflow conditions are above the critical point, which makes use of tabulated properties easier because no discontinuities are encountered. The fluid similarity method used by SNL, if applicable to the current study's conditions, provides a practical way to handle the real-gas implementation without changing the design and off-design equations and avoiding the potential problems with using discontinuous property data.

DETERMINING THE METHODOLOGY FOR THE DESIGN AND PERFORMANCE MAP GENERATION

Before starting an off-design analysis, the design point and the basic geometry of the compressor must be known. The design point is the anchor of the off-design analysis, and calculations provide impeller dimensions, blade lengths, vane and diffuser geometry, number of blades,3w and flow velocity triangles. The velocity triangles determine the blade inlet and exit flow angles and velocities. They are derived from trigonometric relationships and experimental correlations [3, 5, 9].

The design of a radial (or centrifugal) compressor typically uses mean-line analysis, and the design parameters obtained at this step are typically enough to generate the performance maps [3, 5]. A 2D mean-line design code was developed using the design equations and methodology from Mattingly [3], Aungier [9], and Musgrove [5]. The design code requires input for the compressor inlet flow conditions, the shaft speed, and inlet flow angle as well as the inflow thermodynamic conditions such as temperature and pressure. Then the procedure uses trigonometric relations, isentropic flow equations and correlations to calculate the blade inlet and exit flow speeds and their radial and tangential direction components. Two sets of flow coordinates are used: stationary frame and relative frame. The relative frame is a coordinate system attached to the rotor blades and rotates with the shaft and used in calculating the rotor related turbomachinery parameters such as blade flow angles and flow speeds. The slip factor, which indicates the percentage of the blade rotational speed transferred to the fluid flow, is used as an input to the design code. With the calculated blade flow angles from the code, the number of blades and exit relative flow angles are calculated.

The flow Mach numbers at each compressor station are calculated using isentropic flow relations. One important criterion applied in the design codes is to ensure that the calculated design impeller relative exit Mach number does not cause stalling of the compressor at the design point. The calculations move in the direction of the flow, and diffuser and vane sizing are done once the impeller shape and flow specifications are finalized. Flow coefficient-based correlations from Aungier [9] are used in determining the diffuser geometry and flow angles as well as for the vane design.

Typical performance maps for compressors are pressure ratio and enthalpy change or isentropic efficiency versus the mass or volumetric flowrate. It is also very common to use the head vs. flow coefficients, which are nondimensionalized forms of the head and volume flow rate. From a literature review of several compressor design studies [4, 8, 11, 13, 14, 15], there are four possible ways to generate the off-design maps listed as follows.

The first method is to use some correlation equations for pressure ratio and efficiency in terms of inlet or exit flow coefficient or inlet flow conditions. This is the method that requires the least computational effort. The biggest drawback to using this method is that it is specific to a certain flow type and can even be specific to a certain compressor design.

The second method is to use 2D mean-line analysis, where detailed compressor loss models are included with simple velocity triangle calculations to account for the changes in the flow field when the compressor is operated under off-design conditions. However, mean-line analysis methods use ideal gas assumptions and application of real-gas might require equations to be changed.

The third method is to use CFD, where 3D flow effects can be captured with accurate turbulence models. As with every CFD analysis, the software requires an analysis geometry for meshing and is, therefore, highly dependent on the geometry. Unless the geometry is finalized, this method is harder to implement because of the dependence on the analysis geometry. This requires more effort and is typically done only in a detailed design process.

Experiments, which is the method original equipment manufacturers use to generate the maps for their customers, is the most realistic map generation method, and real gas physics are already captured in the results. However, this method not only requires a finalized design but also requires an actual compressor for which multiple tests need to be run and testing equipment needs to be calibrated. The tests need to be run multiple times to ensure data errors are minimized.

Finally, NASA developed the CCODP code for centrifugal (radial) compressors that uses a 2D mean-line analysis and includes analytical enthalpy loss models due to common compressor losses in the off-design operating conditions. The code includes inlet guide vane, inducer incidence, blade loading, skin friction, disk friction, recirculation, vaneless diffuser area, and vaned diffuser loss models. The impact to the overall compressor efficiency is calculated by summing the discussed losses and subtracting them from the change of enthalpy across the compressor under the design conditions. Each loss term takes different values under different inflow conditions, shaft speeds, and mass flowrates, which builds the off-design map curves [12]. What happens to the compressor performance in the off-design is the result of pressure losses at the impeller and vanes, increased flow leakages from blade tips, changes in the disk friction, and diffuser losses. These effects occur simultaneously and result in deviation from the design pressure ratio and efficiency. Compressor losses exist at the design point as well, but they are typically minimized by optimizing the design. The off-design maps are calculated by translating these loss effects into performance metrics.

The 2D mean-line analysis method was selected to develop performance maps because it can handle design changes, real-gas effects, and loss models and is detailed enough for the current level of the design specifications. Therefore, NASA's CCODP code will be used to generate the performance maps of the MC, and a fluid similarity method, which is also applied in a study by SNL for compressor design [11], will be used to find an ideal similar fluid to MC's sCO_2 flow conditions. Then the results from the CCODP code for the similar fluid will be converted to sCO_2 conditions using scaling laws.

USING THE FLUID SIMILARITY METHOD FOR MAIN COMPRESSOR DESIGN AND OFF-DESIGN ANALYSIS

Fluid similarity is frequently used in experiments to replace the actual fluid with an alternate fluid due to availability, safety, or cost. However, fluid similarity principles can also be used in simulations and models for easier modeling [16]. For the current study, the similarity is used to have an ideal and calorically perfect gas that is similar to sCO_2, which would allow use of the design and off-design tools and programs that are developed for ideal gases.

The methodology for the fluid similarity is analogous to using Laplace or Fourier transforms for differential equations, in which these equations are solved in an alternate domain where the solutions are simple. The solutions are then converted back to the actual domain using certain transformation methods. In fluid dynamics, if the actual fluid meets several conditions, then the flow can be represented with another fluid. In the similar fluid domain, the flow is simple and design and experiments can be carried out easier. Then using several scaling laws, the results from the similar fluid can be converted to actual fluids domain [16].

To apply fluid similarity for an application, three conditions should be met: dynamic similarity, geometric similarity, and kinematic similarity. If these conditions are met, then the actual flow can be safely represented with an alternate flow [7, 16].

Dynamic Similarity

The dynamic similarity necessitates that several flow properties should be matched in the similar fluid [7, 16]. In order to have dynamic similarity, the flow Reynolds numbers should be equal. The Reynolds number can be calculated with Equation (6).

$$Re_D = \frac{\rho V D}{\mu} \tag{6}$$

In Equation (6), ρ is the fluid density, V is the flow speed, D is the inlet flow annulus diameter, and μ is the dynamic viscosity. Having the same Reynolds number with the actual fluid would yield identical inlet and exit flow angles with the similar fluid [7]. This would also partially satisfy geometrical similarity as the flowrate and flow angles affect the compressor dimensions directly.

Kinematic Similarity

In order to meet kinematic similarity, the heat transfer properties of the similar fluid should match the actual fluid [7, 16]. Prandtl number, which is the ratio of kinematic viscosity (v) to thermal diffusivity (α), is related to heat transfer characteristics of a fluid, as it is related to the thickness and shape of the thermal boundary layer over the flow surfaces [7, 16]. The Prandtl number can be calculated by Equation (7).

$$Pr = \nu/\alpha \tag{7}$$

In order to have full kinematic similarity, the head coefficient and specific speed of the compressor should also be matched with the similar fluid [7]. The head coefficient, which is used to determine the machine type (previously), is an important measure for compressor performance that relates to the compression ratio. Matching the head coefficient will ensure that the forces and pressures acting on the turbo component will be similar to the actual flow. The specific speed, which is another parameter used in determining the machine type (previously), should also be matched in the similar fluid design to ensure that the flow coefficient and velocity triangles calculated will be similar to the actual flow. If the design of the similar fluid satisfies the above criteria, then the geometric similarity is also assured [7].

Geometric Similarity

The geometric similarity can be ensured when the following properties in the similar fluid are matched: flow coefficient, volumetric flowrate, shaft speed, and Reynolds number. In particular for a compressor, matching the flow coefficient, volumetric flowrate, and shaft speed would make specific speed (see Equation (1)) equal in the similar fluid domain. Having the same design head coefficient in the similar fluid with the actual fluid would make the flow angles similar. These equalities necessitate many of the dimensions be proportional with the actual fluid case, where the transition for dimensions from similar fluid to actual fluid uses several turbomachinery sizing laws, which will be discussed later.

Finding of Similar Fluids for sCO_2

Theoretically, numerous fluids that can satisfy the dynamic similarity (through matching the flow Reynolds number) for the MC sCO_2 flow conditions might exist; however, not all the fluids can match kinematic similarity at the same time. Another important factor in determining the similar fluid is obtaining the fluid's properties from verified fluid databases.

A study by Pucciarelli et al. [17] researched several fluids to replace sCO_2 with an alternate fluid for experimental setups. The various fluids have properties in well-known databases and are also easier to manage than sCO_2. Prandtl number behavior of the fluids was one of the items used to down select potential similar fluids. Ammonia (NH_3), air, Freon or Refrigerant 23 (CHF_3), and water were identified as the best candidates for similar fluid application for CO_2.

The similar fluid for the application of this study is found using the fluid list from Pucciarelli et al. [17]. The flow Reynolds number for the MC CO_2 flow is calculated, which needs to be the same with the actual fluid for the similar fluid. In the flow matching process, the volumetric flowrate of the similar fluid is made equal to the actual fluid to ensure the flow velocities are equivalent. Then REFPROP [2] is used with each candidate fluid to find the matching flow conditions with the sCO_2. The main criteria checked for the matched conditions include being an ideal gas, a calorically perfect gas, and in a vapor or supercritical phase at the matched conditions. Based on these criteria, water and ammonia were eliminated due to being in a liquid phase and/or non-ideal at the matched inflow conditions.

Air and Freon were identified as possible similar fluids for our application. The properties of these fluids at the matched conditions are compared in Table 4.

Table 4 Similar fluids properties compared at the matched conditions

Parameter	sCO_2	Air	CHF_3 (Freon)
Pressure [psia]	887	3550	5510
Temperature [F]	71.44	22	260
Compressibility Factor	0.144	1.03	0.95
Specific Heat Ratio	4.479	1.197	1.56
Density [lbm/ft^3]	47.285	19.249	54.291
Mass Flowrate [lbm/s]	6383	2599	7329
Volumetric Flowrate [ft^3/s]	135	135	135
Reynolds Number	65.7E6	65.7E6	65.7E6

From a comparison of the values, air mass flowrate at the matched conditions is significantly different from CO_2 due to the higher difference in density than Freon. Consequently, using air as the similar fluid would cause significant geometrical differences, which might result in problems when scaling the maps. Freon, which is a common refrigerant, has a mass flowrate very close to the sCO_2 case, which would ensure that the designed geometry with this fluid would be very similar to the CO_2 case. This is mainly because the density of Freon is very close to CO_2, making the scaling of the maps easier and with fewer errors. Therefore, Freon was selected over air as the similar fluid for the current study.

Map Scaling for Fluid Similarity

Turbomachinery scaling or affinity laws are used frequently in industry for performance map scaling of similar compressors or pumps. If the two pumps or compressors are geometrically similar in terms of the design, which means either the proportions or dimensions of the two machines are similar to each other, then the performance map of one machine can be used to estimate the performance of the other. Scaling laws can be used to scale different fluid operation cases, because they include scaling factors for flow properties. In general, affinity laws are used to scale the pressure ratio, head coefficient, isentropic efficiency, or work input. In this context, we use these scaling laws to scale maps from the Freon compressor design to sCO_2 compressor.

Two types of performance curves are generated in this study: pressure ratio versus mass flowrate and isentropic efficiency versus mass flowrate. Therefore, only the scaling of these two parameters will be discussed in this section. From the fluid similarity principles, the head coefficient of the sCO_2 compressor will be equal to the Freon compressor at all operation points. Therefore, a pressure ratio scaling formula can be derived by equating the head coefficient formulae (see Equation (2)), which is given in Equation (8).

$$\pi_c = \left\{1 + \frac{z_{CHF_3}\gamma_{CHF_3}R_{CHF_3}T_{CHF_3}}{z_{CO_2}\gamma_{CO_2}R_{CO_2}T_{CO_2}} \frac{\gamma_{CO_2-1}}{\gamma_{CHF_3-1}} \left[\pi_C^{*(\gamma_{CHF_3}-1)/\gamma_{CHF_3}} - 1\right]\right\} \tag{8}$$

In Equation (8), z is the compressibility factor, R is the gas constant, T is the total temperature, and γ is the ratio of specific heats. Equation (8) is used for each point in the Freon compressor pressure ratio (π_C^*) versus mass flowrate map curve to transform the map data from Freon to sCO_2 values.

The mass flowrate for sCO_2 is scaled from Freon using the density ratio of two fluids. From the fluid similarity principles, the flow speed and inflow area of the sCO_2 compressor will be similar to the Freon compressor and, therefore, the only scaling factor in the equation is the fluids' densities. The mass flowrate scaling formula is given with Equation (9).

$$\dot{m}_{CO_2} \approx \frac{\rho_{CO_2}}{\rho_{CHF_3}} \dot{m}_{CHF_3} \tag{9}$$

The scaling formula for the isentropic efficiency is obtained from turbomachinery affinity laws, typically applied to estimate the performance of a compressor that has dimensions similar to another compressor of which its performance maps are known [7, 18]. The scaling formula for the isentropic efficiency is given with Equation (10).

$$\frac{1-\eta_{CHF_3}}{1-\eta_{CO_2}} \approx \left(\frac{d_{2,CO_2}}{d_{2,CHF_3}}\right)^n \tag{10}$$

Equation (10) requires the ratio of impeller diameters of two compressors and an exponent n. Although the Freon compressor is geometrically similar to the sCO_2 compressor, the impeller diameters are not exactly equal. However, the specific speeds of the two compressors are equal per the fluid similarity principles. Using this design feature and the wheel speed equations, a formula for the diameter ratio can be obtained as given in Equation (11).

$$\left\{\frac{h_{t1}\left(\pi_c^{(\gamma-1)/\gamma e_c} - 1\right)}{\varepsilon}\right\}_{CO_2} = \frac{d_{2,CHF_3}}{d_{2,CO_2}} \left\{\frac{h_{t1}\left(\pi_c^{(\gamma-1)/\gamma e_c} - 1\right)}{\varepsilon}\right\}_{CHF_3} \tag{11}$$

After calculating the diameter ratio from Equation (11), the exponent n of Equation (10) can be found using the design point efficiencies of the Freon and sCO_2 compressors, respectively. It should be noted that the given form of Equation (10) is an approximate form, and the exact equation can only be obtained using a multiplication factor obtained with experimental data for the actual compressor for the isentropic efficiencies.

VALIDATION OF THE METHODOLOGY

Although the fluid similarity method has been previously used and validated by SNL [11], there are some variations in the application of the methodology for this current study, mainly in the map scaling.

The validation procedure starts with finding a similar fluid to sCO_2 at the SNL compressor's inflow conditions. The inflow temperature is 305.3K and pressure is 7.687 MPa for this compressor. The design point mass flowrate is 3.53 kg/s, design shaft speed is 75000 rpm, and the design pressure ratio is 1.8. The design isentropic efficiency is 80% [11]. The design flow conditions correspond to a point slightly above the critical point, but REFPROP calculations show that the thermodynamic properties, such as the specific heats and the specific heat ratios, undergo rapid changes that necessitate use of the fluid similarity method to be able to use design and off-design methods developed for calorically perfect gases.

For the given inflow conditions of the SNL compressor, the flow Reynolds number is calculated to be 148.8E6, isentropic head is calculated to be 5002 ft.lbf/lbm, and specific speed is calculated to be 0.97. The fluid list for similar Prandtl number behavior with sCO_2 is used from the study by Pucciarelli et al. [17] to find a similar fluid. Investigation of the fluids with REFPROP analysis show that only air is suitable as a similar fluid at the given design conditions, whereas Freon is either a liquid or not ideal at matched conditions. Other refrigerant fluids were not tried as the methodology is independent of the specific type of the similar fluid used as long as the dynamic, kinematic, and geometric similarities are satisfied.

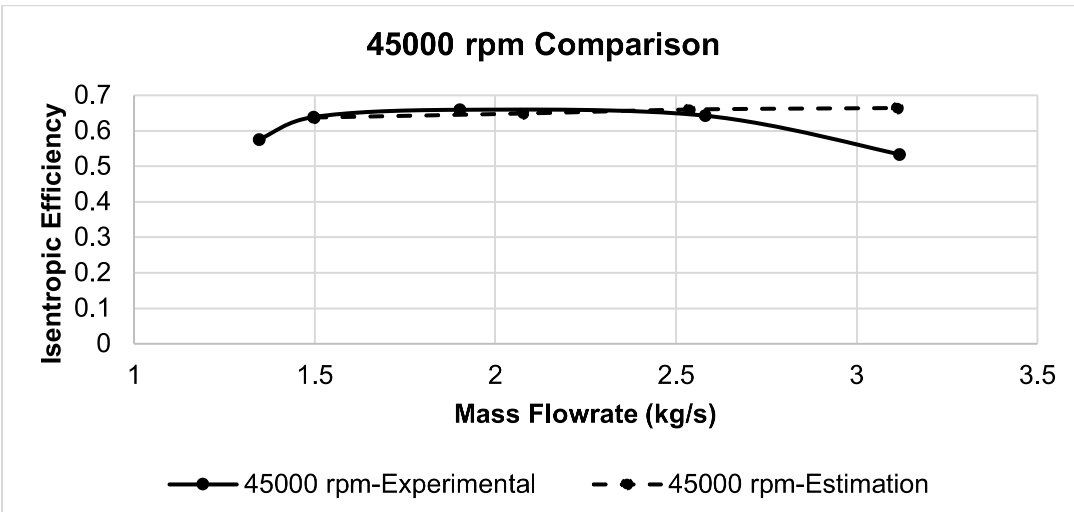
The matched similar fluid inflow conditions for air are identified to be at 216.48K and 28.99 MPa. The equivalent compressor design pressure ratio calculated by Equation (8) is obtained as 1.255. The density of air at these conditions is 457.55 kg/m^3, while sCO_2 density is 585.95 kg/m^3. The difference in densities between the actual and similar fluid is slightly higher than the difference in densities for the current study's fluid similarity methodology. With the indicated similar fluid conditions, the design for air compressor uses the in-house design code. Equation (11) is then used at the design point to calculate the impeller diameter scaling from air to sCO_2 compressor. The diameter ratio is then used to scale other dimensions calculated for the air compressor to sCO_2 and compared with SNL compressor data in Table 5. Note that per the fluid similarity principles, the flow angles will be the same for both air and sCO_2 compressors; therefore, no scaling is needed for flow angles.

Table 5 Comparison of the calculated compressor design parameters with SNL compressor data

Parameter	**Current Study**		**SNL [11]**
Flow	Air	sCO_2	sCO_2
Impeller Diameter, d_2	0.068 m	0.048 m	0.051 m
Impeller Length, L	0.03 m	0.02 m	0.02 m
Inlet Tip Diameter, R_{1t}	0.01 m	0.008 m	0.009 m
Impeller Inlet Angle, β_1	57^0	57^0	50^0
Impeller Exit Angle, β_2	49^0	49^0	50^0
Throat Diameter, b_2	0.001 m	0.0007 m	0.0008 m
Exit Flow Angle, α_3	72.6^0	72.6^0	71.5^0
Number of Blades	13	13	12

The differences in dimensions are expected as the design procedure applied by SNL not only includes main-line analytical calculations but also some calculations that include detailed turbomachinery analysis [11]. However, the flow angles calculated with the current study are very close to the compared data, indicating that the obtained design parameters can be used to conduct the performance analysis. The NASA CCODP code is used with the air compressor design parameters to obtain the performance maps for pressure ratio and isentropic efficiency for off-design conditions. SNL provided experimental off-design measurements for their compressor at 73%, 67%, and 60% of design shaft speed that correspond to 55000, 50000, and 45000 rpm, respectively. Experimental data were used by SNL to validate their performance map predictions with their analytical models and fluid similarity method. The measured efficiency data from the compressor performance tests include data from transients caused by closing the MC valve, and these data are not included in the method validation comparisons of this study [11].

The experimental data from indicated compressor efficiency tests in SNL [11] and the estimation by the current study are compared in Figure 5.



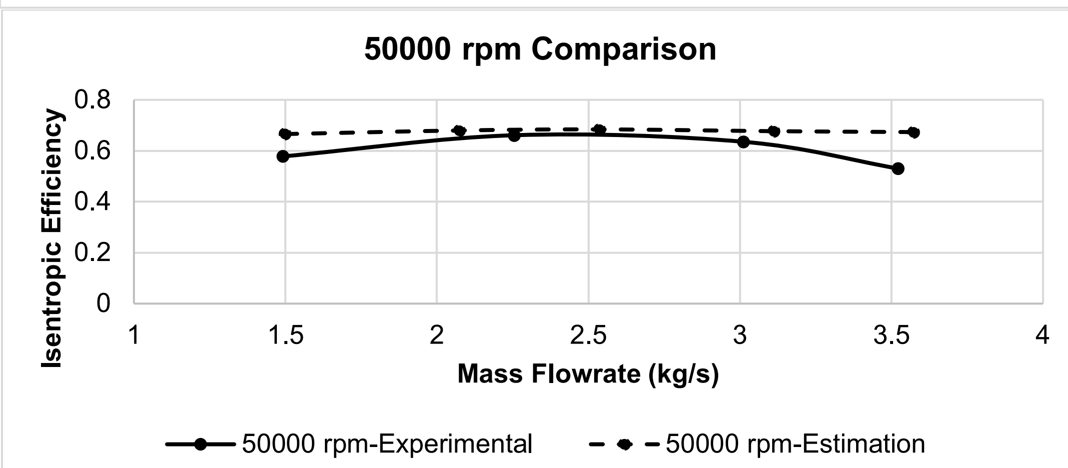


Figure 5 Comparison of isentropic efficiency estimation with current study's fluid similarity method with experimental data

The estimations with the current study's methodology show good agreement up to around 2.5–3 kg/s flowrates. The discrepancy at higher mass flowrates is explained by the increased impact of the windage losses as the leakage flow to the seal also increases when the mass flowrate is increased in the test compressor [11]. The current model using CCODP does not have a windage loss model for the windage losses between the rotor (impeller) and stators (vanes), while the SNL model includes these losses on top of the disk friction losses already calculated by CCODP. Windage losses are estimated to be more pronounced for the shaft speeds higher than 20000–25000 rpm [15] and are estimated to be negligible for the MC design conditions of this study.

DESIGN ANALYSIS

Per the fluid similarity principles, the head coefficient and specific speed for the sCO_2 and Freon flow cases should be equal. The volumetric flowrates for the two fluids are considered equal, which necessitates the shaft speed to be equal for both cases in order to have the same design specific speed. With the indicated conditions, there is an equivalent design compressor pressure ratio for the Freon case, which is different than the pressure ratio for the sCO_2 case. The value of this equivalent pressure ratio can be found by Equation (2) for both fluids and equating them to solve for the equivalent pressure ratio for the Freon case. The pressure ratio and related results with Freon are then scaled back to CO_2 values with appropriate scaling methods. The Freon equivalent pressure ratios of both stages are compared with actual fluid values in Table 5.

Table 6 Calculated design pressure ratios for the Freon

Stage No	sCO_2	Freon
1	2.506	1.445
2	2.484	2.013

The Freon compressor design matches the Reynolds number for the same head coefficient and specific speed with CO_2 flow conditions. The in-house design code is used to design the compressor stages using mean-line design principles from Mattingly [3] and Aungier [9]. Thermodynamic properties of Freon are used in the design code, where different property sets are used at the compressor inlet, impeller inlet, and impeller exit locations to improve the accuracy. No changes to the design equations are made as Freon is an ideal and calorically perfect gas at the matched conditions. A feasible design uses several industrial design practices and a Baije chart.

At the beginning of the design process, a Baije chart was used to find some design parameters such as the slip factor, shaft speed, and specific diameter to have a design efficiency around 75–85 percent. The shaft speed selection is important in design as it affects all of the design parameters. Two options exist for the MC in terms of the shaft speed:

- Fixed at 3600 rpm, which matches the generator frequency and could be advantageous for turbine coupling
- Greater than 3600 rpm to ensure compressor isentropic efficiency in the range of 75–85 percent

Two preliminary designs for each shaft speed case are made. These designs use the Baije chart and the in-house design code. The general design specifications of these two options are compared in Table 6.

Table 7 Comparison of the design options for different shaft speeds

Parameter	3600-rpm design	Highest Efficiency
Rotational Speed	3600	16500
Impeller Diameter	31.5”	15.5”
Slip Factor	0.85	0.8
Polytropic Efficiency	0.8–0.83	0.84–0.87
Configuration	6-Stage Compressor with Intercooling (I-C) (3 Stages+I-C+3 Stages)	2-Stage Compressor with I-C (1 Stage+I-C+1stage)

For the 3600-rpm shaft speed, the impeller diameter increases significantly, and requires 6-stage compression to reach the cycle design specifications. This design is also projected to have lower efficiency than higher rpm designs. When the design shaft speed is increased to 16500 rpm, the impeller diameter reduces to manufacturability standards, and the compression required by the cycle can be completed with 2 stages. Based on the projected efficiency, number of stages, and impeller diameter, the higher shaft speed design is selected over the 3600-rpm design.

The selected design is then refined by applying several industry practices for better operation. Several design standards from Aungier [9], Wiesner [19], and Walsch et al. [20] are adapted. Some of the applied design criteria from the indicated literature are listed in Table 7.

Table 8 Applied design criteria in the design code

Parameter	Design Criteria
Mean Inlet Mach Number	0.4–0.6
Impeller Backsweep Angle	$<40^0$
Inducer Hub-to-Tip Ratio	0.35–0.5
Exit Flow Speed	<300 m/s
Impeller Diameter	<0.8 m
Exit Flow Mach Number	<0.25
Slip Factor	0.8–0.95

Meeting the listed standards from Table 7 in the design ensures that the design meets industrial design practices. For example, the impeller diameter criteria come from a manufacturability limit and stress limits on typical radial compressor blade materials [20]. Most of the listed criteria are determined by experiments, design field experiences, and manufacturability limitations and are also good for reducing compressor losses. With the indicated design procedure, the compressor design parameters are determined as listed in Table 8.

Table 9 Main Compressor Design Parameters for each stage

Parameter	Stage 1	Stage 2
Inlet Hub Diameter, R_{1h}	2.64”	2.39”
Inlet Tip Diameter, R_{1s}	5.28”	4.79”
Impeller Radius, R_2	7.81”	9.72”
Blade Length, L	4.8”	8”
Impeller Inlet Angle, β_1	38^0	37^0
Impeller Exit Angle, β_2	32^0	28^0
Slip Factor	0.81	0.86
Number of Impeller Blades	9	15
Throat Diameter, b_2	0.46”	0.22”
Pressure Ratio	1.445	2.013
Isentropic Efficiency	0.78	0.85
Flow Coefficient	0.09	0.04
Impeller Exit Mach Number	0.65	0.9
Stage Exit Mach Number	0.2	0.2

The dimensions presented in Table 8, correspond to the indicated locations on the compressor in Figure 5.

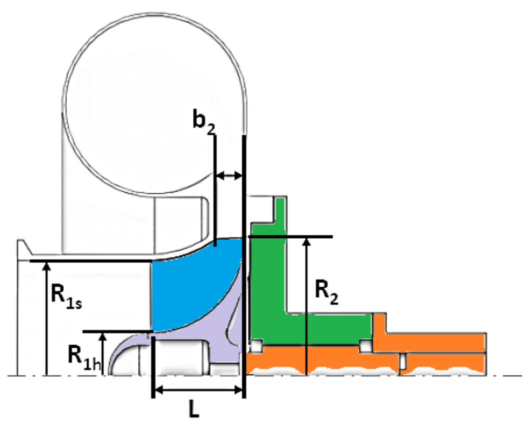


Figure 6 Calculated compressor dimensions in the design procedure

After using the 2-stage design with the specifications from Table 8 to generate the off-design maps with CCODP, the Stage 2 operational range was found to be too narrow (i.e., Stage 2 enters to surge below 70% of design mass flowrate). The high relative impeller design tip Mach number, 0.9, is indicative of this behavior. Having a higher relative impeller tip Mach number increases the stage efficiency [20] but might also reduce the operational range because the flow speed increases beyond the sonic speed for lower inlet flowrates. This would cause flow separation on the blades.

Based on the problems in the off-design analysis, Stage 2 needs to be re-designed; two possible options can resolve the discussed issues:

- Divide Stage 2 into two equal stages, which would reduce the pressure ratio and impeller Mach number of each stage
- Run Stage 2 at lower rpm than the first stage, which would require an internally geared compressor design. In this case, the second stage will not be on the same shaft with the first stage

The first option was selected in order to reduce the design complexity, possible turbine coupling options from the cycle, and to avoid additional losses that will be introduced with integral gearing.

The design procedure is repeated for the second and third stages, each having the same pressure ratio, comprising three compressor stages in total for the MC, but all stages are running on a common shaft at 16500 rpm design speed. Intercooling is applied between the first and second stage only, with no cooling between the second and third stage. Fluid similarity with Freon is used to design and perform the off-design analysis of the last two stages. The design parameters of the 3-stage compressor are presented in Table 9.

Table 10 Updated main compressor design parameters for each stage

Parameter	Stage 1	Stage 2	Stage 3
Inlet Hub Diameter, R_{1h}	2.64”	2.39”	2.46”
Inlet Tip Diameter, R_{1s}	5.28”	4.79”	4.91”
Impeller Radius, R_2	7.81”	6.57”	7.32”
Blade Length, L	4.8”	3.9”	4.7”
Impeller Inlet Angle, β_1	38^0	37^0	38^0
Impeller Exit Angle, β_2	32^0	27^0	26^0
Slip Factor	0.81	0.8	0.84
Number of Impeller Blades	9	9	13
Throat Diameter, b_2	0.46”	0.52”	0.39”
Pressure Ratio (sCO_2 Equivalent)	2.5	1.583	1.583
Isentropic Efficiency	0.78	0.89	0.91
Flow Coefficient	0.09	0.13	0.09
Impeller Exit Mach Number	0.61	0.65	0.64
Stage Exit Mach Number	0.2	0.22	0.19

OFF-DESIGN (PERFORMANCE) ANALYSIS

Once the designs of the stages are completed, required inputs for the off-design analysis are obtained. The NASA CCODP code [12] is used with the presented design data to generate the off-design maps for the Freon compressor. Since Freon is an ideal and calorically perfect gas at the matched conditions, no changes need to be made to the performance map calculation code. Maps are generated for seven different shaft speeds, starting from the design speed down to 40 percent of the design shaft speed. The lowest value of the shaft speed is determined by investigating the pressure ratio values obtained at each speed. Pressure ratio and isentropic efficiency curves are generated for each speed and performance maps are generated. Then the maps are scaled from Freon to sCO_2 using the methodology described earlier.

For the first stage, the performance maps for the pressure ratio and the isentropic efficiency are obtained as presented in Figure 6. The surge point of the first stage is at 32% of the design mass flowrate (2893 kg/s).

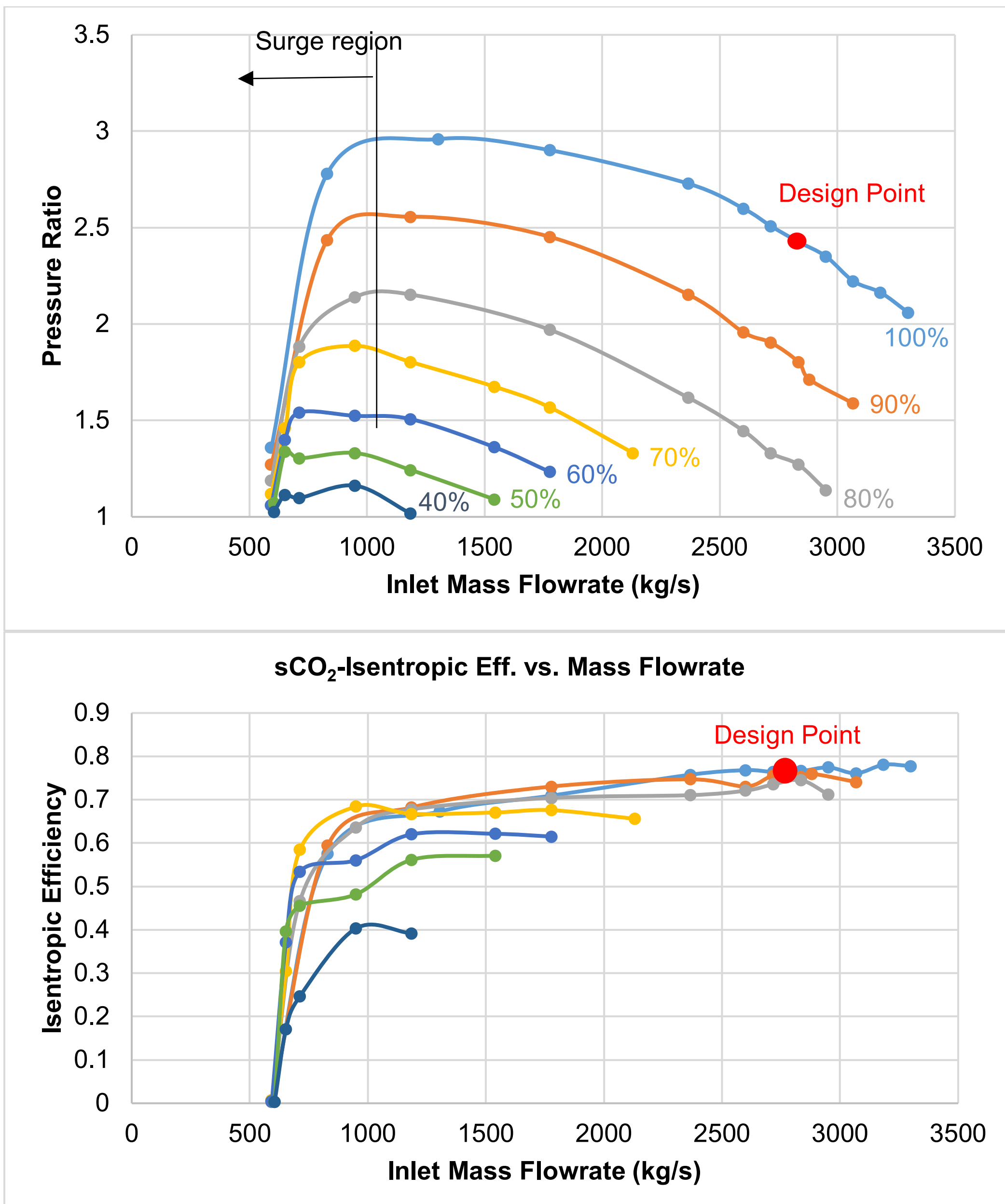


Figure 7 Calculated compressor performance maps for Stage 1 (after scaling for sCO_2)

The primary observation from the obtained curves is that the pressure ratio curves are wide parabolic curves. The shape of these curves is a hybrid of typical pump and compressor maps.

The efficiency curves are highly similar to the typical pump efficiency curves [18]. The resemblance of the performance maps to pump curves is an expected trend because of the trans-critical inlet conditions.

The reduced operational range of Stage 2 at the design shaft speed is increased when the pressure ratio is reduced. CCODP calculated no compression from the compressor for the shaft speeds below 40 percent design rpm and, therefore, this value corresponds to the lowest possible speed in the performance curves. The performance curves for Stage 2 are shown in Figure 7.

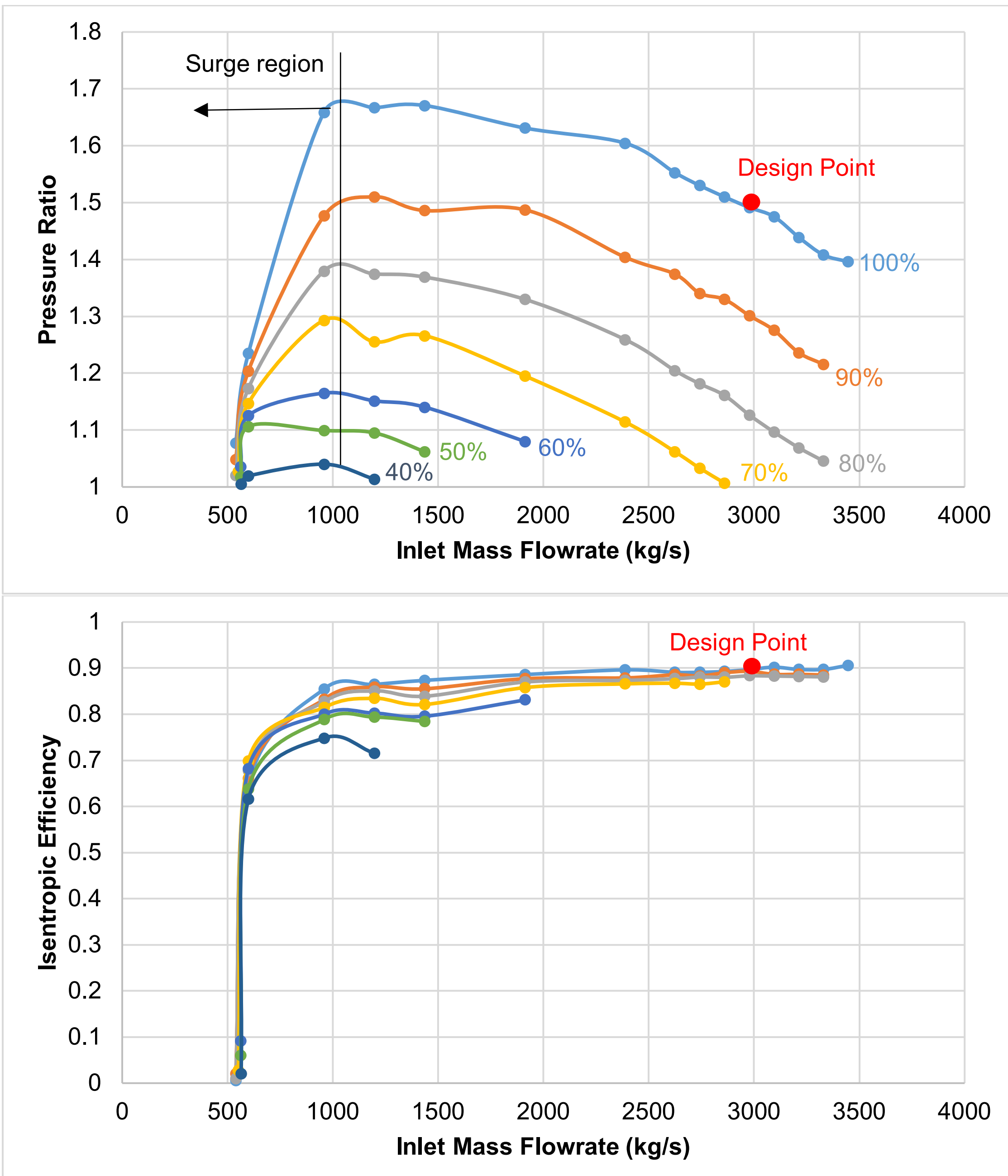


Figure 8 Calculated compressor performance maps for Stage 2 (after scaling for sCO_2)

At the design point, the third-stage inlet conditions are calculated to be supercritical, causing this stage to have the highest efficiency among the other stages. At the design point, the first-stage isentropic efficiency is 78%, second-stage efficiency is 89% and third-stage efficiency is 91%. Stage 1 efficiency is the lowest because of operating at the two-phase inlet conditions, the second stage has higher efficiency due to the intercooling, and the third stage has the highest efficiency as it has supercritical inlet conditions. The performance curves for the third stage are shown in Figure 8.

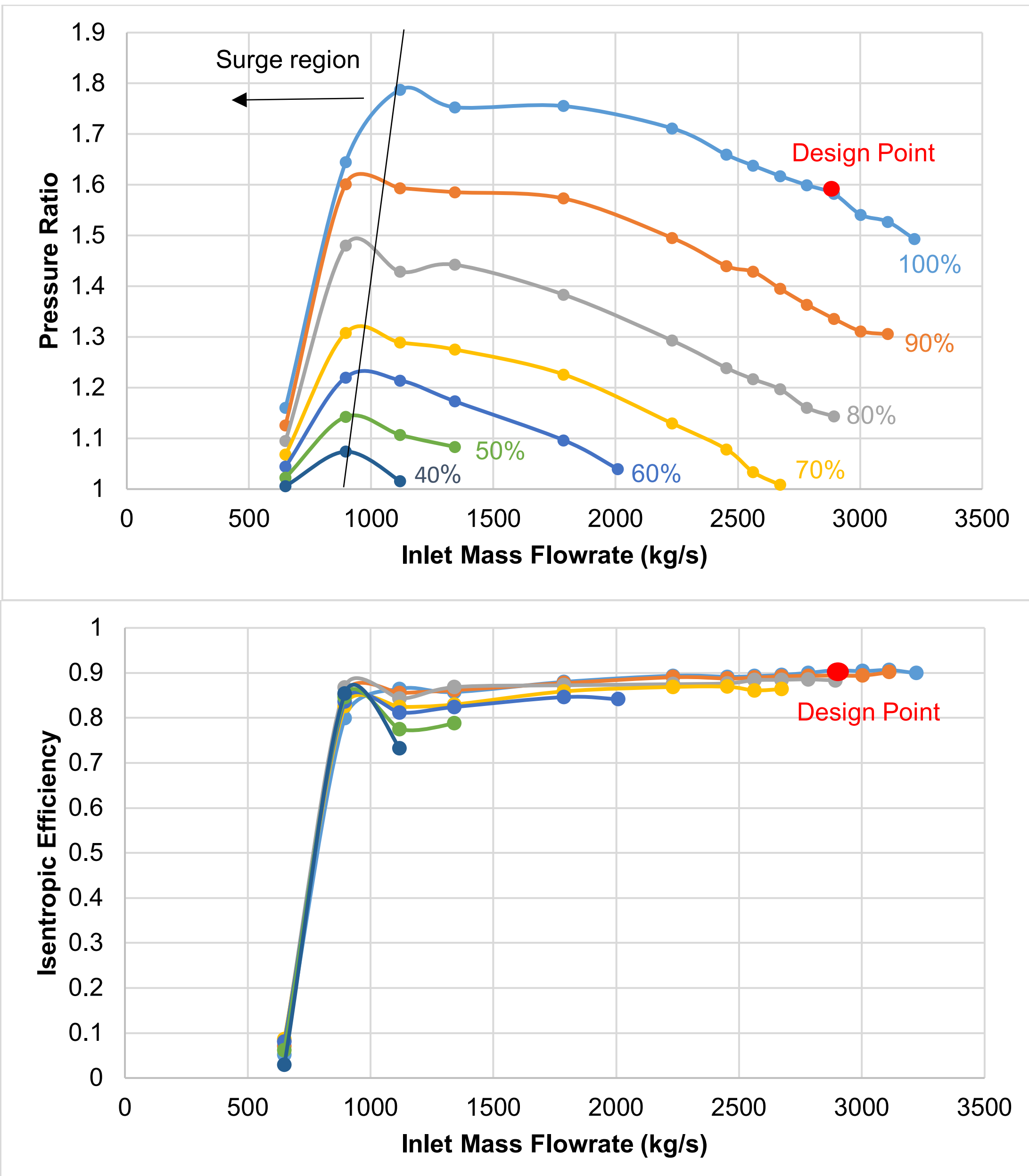


Figure 9 Calculated compressor performance maps for Stage 3 (after scaling for sCO_2)

For all stages, the reduction in efficiency with the reduced shaft speed becomes more evident below 60% design shaft speed. For the 40% design shaft speed, the efficiency drop is very steep, and the compressor is operable for a very narrow region as compared to other shaft speeds. The reduction in efficiency and the operational range with the reduced shaft speed is expected because of the higher mismatch between the actual impeller diameter and optimum impeller diameter for the reduced flow conditions. The decrease in the pressure ratio with the increasing mass flowrate is comparably higher for Stage 2 and Stage 3 than the first stage, probably because of the lower liquid fraction at the inflow for these two stages.

CONCLUSION

A radial (centrifugal) compressor is designed for the MC component of an indirect sCO_2 cycle, where the inflow conditions are trans-critical. The thermodynamic properties of the flow at the inlet conditions are non-ideal, and calorically perfect gas assumptions will not be valid due to high gradients in specific heat capacity values. Indicated flow conditions require using real gas analysis for the design and off-design calculations. Inclusion of the real gas analysis in the design and off-design calculations would require changing the equations and, most importantly, would require the use of tabulated thermodynamic properties with high gradients, which will eventually yield to problems if they are used in interpolations.

The fluid similarity method is used to design and generate performance maps of the compressor. For the MC's flow conditions, Freon is determined to be the similar fluid, which is both ideal and calorically perfect at the matched conditions. The CCODP code by NASA [12] is used to generate the performance maps for the Freon compressor, which are then scaled for the sCO_2 application.

Calculated maps show centrifugal pump-like characteristics for the pressure ratio and efficiency. The lowest shaft speed calculated is 40% of the design speed (6600 rpm), as the code predicts no compression below this value. The first-stage efficiency is calculated to be the lowest, followed by the efficiency of the second stage, with the third stage having the highest efficiency. The reasoning behind this pattern is the liquid-like behavior of the first-stage inflow, and the existence of the intercooler to increase the efficiency of the second stage. The third stage operates at supercritical conditions, where efficiency is higher. For all stages, the reduction in efficiency with the reduced shaft speed becomes more evident below 60% design shaft speed. The value of the 100% design shaft speed is, therefore, important in many aspects of compressor performance. In the current design made for the main compressor, the design shaft speed of 16500 rpm is selected over 3600 rpm to achieve higher efficiency and to reduce the number of compression stages.

In general, the calculated performance map data shows that surge margin is narrower for sCO_2 compressors than for air compressors, and small changes in the mass flowrate can impact sCO_2 compressor performance more rapidly. The reduced surge margin impacted the initial design of the Stage 2, which initially enters the surge zone as early as at 70% of the design mass flowrate. The operational range of the second stage is increased after dividing this stage into two equal stages.

The generated performance map data will be used in ASPEN Plus software for use in off-design and dynamic cycle simulations. The fluid similarity method and the obtained performance map data will also be used to generate the dataset for inlet guide vane control option for the MC.

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DISCLAIMER

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