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Direction for High-Performance Supercritical CO₂ Centrifugal Compressor Design for Dry Cooled Supercritical CO₂ Brayton Cycle

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Abstract: To overcome the degradation of the cycle efficiency of a supercritical carbon dioxide (S-CO₂) Brayton cycle with dry cooling, this study proposes an improved design of an S-CO₂ centrifugal compressor. The conventional air centrifugal compressor can achieve higher efficiency as backsweep angle increases. However, the structural issue restricts the maximum allowable angle (−50°~−56°). In this study, an S-CO₂ centrifugal compressor performance was examined while changing the backward sweep angle at impeller exit to study if the previous optimum backsweep angle for an air centrifugal compressor is still valid when the fluid has changed. It is shown through an analysis that an S-CO₂ centrifugal compressor can achieve the highest efficiency at −70° backsweep angle, which is greater than the typical design value. The S-CO₂ centrifugal compressor is less restricted from a structural integrity issue because it has low relative Mach number regardless of the low sound speed near critical point ($T_c = 304.11$ K, $P_c = 7377$ kPa). It is also shown in the paper that the variation of compressibility factor does not impact on its total to total efficiency since its Mach number is still lower than unity. Finally, it is also shown that a backward sweep impeller can achieve higher pressure ratio and operate stably in wider range as the mass flow rate is decreased. As further works, the suggested concept will be validated by the structural analysis and the compressor performance test.

Keywords: S-CO₂ Brayton cycle; Dry cooling system; S-CO₂ centrifugal compressor; high-performance compressor; large backsweep angle

1. Introduction

A supercritical CO₂ (S-CO₂) Brayton cycle is a promising power technology for the next generation heat to power conversion systems due to its high cycle efficiency at moderate turbine inlet temperature (550~750 °C), compact cycle configuration, and alleviation of turbine blade erosion in comparison with the steam Rankine cycle [1]. Due to these benefits, it has been considered as a future power system for various heat sources, fossil fuel, waste heat, concentrating solar power (CSP), fuel cells and nuclear.

Meanwhile, demands for power have been steadily rising in inland areas without access to abundant water resources. To minimize water consumption a dry cooling system is necessary for the power plants built in these areas. The dry cooling system is to use the atmosphere as an ultimate heat sink for cooling the whole power plant. Typically, the average difference between the dry-bulb temperature in an arid climate and the minimum temperature of the power plant is

as high as 10 °C [2]. The characteristics of a dry cooling system relate to disadvantage of a higher compressor inlet temperature (CIT) than the water cooling system, which results in the power plant efficiency deterioration.

An S-CO₂ Brayton cycle can achieve high cycle efficiency by compressing CO₂ near the critical point ($T_c = 30.98\text{ °C}$, $P_c = 7377\text{ kPa}$) because it significantly reduces the compressor work and maximizes regeneration. Figure 1 shows the cycle efficiency degradation as the S-CO₂ CIT moves above the critical point. Thus, an S-CO₂ Brayton cycle with the dry cooling system inevitably faces substantial decrement of the cycle efficiency. Various attempts have been made to recover the efficiency of the system. Turchi et al. and Conboy et al. studied various cycle layouts and the cycle optimum conditions to overcome the decline of cycle efficiency [3,4]. Zeyghami and Khalili proposed a daytime radiative cooling concept to improve system performance [5]. Ma et al. suggested an S-CO₂ Brayton cycle integrated with LiBr absorption chiller to minimize exergy losses [6].

Unlike previous approaches, this study aims to increase the cycle efficiency of an S-CO₂ Brayton cycle by improving the aerodynamic performance of an S-CO₂ centrifugal compressor. Figure 2 shows classical configuration of an S-CO₂ Brayton cycle. Compared to the conventional Brayton cycles, an S-CO₂ Brayton cycle feature low expansion ratio and high regeneration. Figure 3 shows the sensitivity analysis of S-CO₂ Brayton cycles relative to CIT and compressor isentropic efficiency variations. It can be found that the cycle efficiency becomes more and more sensitive to the compressor efficiency as CIT increases. An S-CO₂ Brayton cycle operating near the critical point is relatively insensitive to the compressor efficiency on the cycle performance because of its small compression work. However, as the cycle moves further away from the critical point and consequently requires more compression work, higher compressor efficiency becomes very important to the system performance. This supports the importance of research for improving the compressor performance in an S-CO₂ Brayton cycle with dry cooling.

When designing turbomachinery, complex analyses to evaluate mechanical feasibility have to be performed, which finally leads to fixing the design parameters. However, the design process often involves fixing design parameters from experience of each designer to close the mathematical relations of design parameters and performance. Various institutions have tried to manufacture and test S-CO₂ compressors in their facilities by collaborating with turbomachinery vendors. Sandia National Laboratory and Barber-Nichols Inc. in the U.S. and KAIST, Korea Atomic Energy Research Institute and Jinsol Turbo Machinery in Korea are examples of such collaboration [7–9]. However, these experimental studies have mostly focused on applying existing accumulated air compressor technologies to the S-CO₂ compressor since an S-CO₂ Brayton cycle is not yet mature power generation technology. More specifically, due to issues with bearing and seal, the amount of data is limited to investigate how different the S-CO₂ compressor can be in terms of aerodynamic characteristics from the air compressor [10–12].

The most important process in the aerodynamic design of a centrifugal compressor is to select the backswEEP angle at impeller exit because it can improve the isentropic efficiency, extend stable operating range and raise the degree of reaction. Traditionally, the centrifugal compressor has pursued higher backswEEP angle, however, due to the limitation on the materials to withstand high levels of tip speed and temperature the angle is restricted to about -50° [13–16]. For designing the S-CO₂ centrifugal compressor, thermodynamic properties variation near the critical point of CO₂ has to be considered unlike from the air centrifugal compressor. Thus, it is natural to investigate how these characteristics may affect the optimum backswEEP angle. The selection of the optimum angle will depend on its operating conditions such as the static temperature and the static pressure at the compressor inlet. Therefore, this paper will cover the effect of backswEEP angle changes on the S-CO₂ centrifugal compressor performance using a 1D mean stream-line method.

Figure 1. Theoretical cycle efficiency of S-CO₂ recuperation Brayton cycle.

Figure 2. S-CO₂ recuperation Brayton cycle.

Figure 3. Performance of S-CO₂ Brayton cycle relative to CIT and compressor isentropic efficiency variations.

2. Methodology

KAIST research team has been working on a compressor design tool for the S-CO₂ compressor design and analysis. The development of KAIST-TMD which is the name of the compressor design tool is motivated to overcome limitations on the conventional tools relying on the ideal gas approach

or simplified real gas approach, since conventional compressor design methodologies may become less accurate due to the behavior of CO₂ near the critical point. A turbomachinery in-house code, namely KAIST-TMD, is utilized in this study. KAIST-TMD and a benchmark compressor are described below.

2.1. KAIST-TMD

There have been many efforts to develop an S-CO₂ compressor design and analysis tool. Most of the researchers utilized the existing commercial codes with minor modifications. For this reason, many codes have convergence issues near the critical point [17]. The main differences between KAIST-TMD and the previous ones are modification of correlations based on ideal gas assumption and integration of real gas property database NIST [18] and the design tool. As a result, the calculation procedure is mainly based on calculating enthalpy and pressure, which is more straightforward and has less error for adopting the definition based static to stagnation conversion method directly. Equation (1), which was commonly used in the previous tools, has a faster calculation speed but it is accompanied by large errors near the critical points where thermodynamic properties change sharply [19]. On the other hand, Equation (2), which is used in KAIST-TMD, has a slower calculation speed but reflects thermodynamic properties of S-CO₂ even near the critical point [20].

$$\frac{P_o}{P_s} = \left(\frac{T_o}{T_s}\right)^{\frac{\gamma-1}{\gamma}} = \left(1 + \frac{\gamma-1}{2}M^2\right)^{\frac{\gamma}{\gamma-1}} \quad (1)$$

$$h_o = h_s + \frac{V^2}{2} \quad (2)$$

Even though the computational fluid dynamics (CFD) is a powerful tool to analyze turbomachinery, the 1D mean streamline analysis method is still a very important tool because it determines key design parameters. Since the flow mechanism in a compressor is complicated, the simplified design method is generally preferred for compressor design process. The 1D mean streamline method on the basis of basic aerodynamic design and empirical correlations is widely applied. It is a design process to determine velocity triangles with Euler turbine equation and continuity equation as shown in Equation (3), Equation (4). Empirical correlations such as loss models and slip factor which cause entropy generation are applied. KAIST-TMD adopts the following 1D mean streamline analysis method.

$$h_{o2} - h_{o1} = U_2 C_{w2} - U_1 C_{w1} \quad (3)$$

$$\dot{m} = \rho(h_s, P_s)AV \quad (4)$$

For vaneless diffuser design, Equations (5)–(8) which are derived from the conservation equations in the vaneless diffuser region were utilized in KAIST-TMD [21]. Figure 4 represents a flow chart of its main algorithm. This tool generates a compressor having the highest isentropic efficiency while meeting the outlet pressure conditions.

$$C_m \frac{dC_m}{dr} - \frac{C_w^2}{r} + C_f \frac{C^2 \cos \alpha}{b \sin \phi} + \frac{dp}{\rho dr} = 0 \quad (5)$$

$$C_m \frac{dC_w}{dr} + \frac{C_m C_w}{r} + C_f \frac{C^2 \sin \alpha}{b \sin \phi} = 0 \quad (6)$$

$$\frac{d\rho}{\rho dr} + \frac{dC_m}{C_m dr} + \frac{dh}{h dr} + \frac{1}{r} = 0 \quad (7)$$

$$\frac{dh}{dr} + C_m \frac{dC_m}{dr} + C_w \frac{dC_w}{dr} = 0 \quad (8)$$

Figure 4. KAIST-TMD main algorithm.

2.2. Loss Models

The selection of loss models is a significant process in the 1D method because it determines the accuracy of analysis results. Martins and Zhu et al. adopted loss models through validation with experimental results and CFD results for an axial turbine and a centrifugal compressor, respectively [22,23]. KAIST-TMD adopts empirical loss models and slip factor to estimate the S-CO₂ compressor performance. The loss models are empirical correlations for irreversibility estimation of turbomachinery. Since KAIST-TMD is validated with the S-CO₂ compressor experimental data, the design results can be believed to have high fidelity under various different design conditions [24]. Table 1 summarizes the selected loss models and Figures 5 and 6 illustrate the mechanisms of each loss.

The internal losses consist of incidence loss, blade loading loss, skin friction loss, mixing loss and clearance loss for the unshrouded impeller. All the internal losses are supposed to contribute to the efficiency as well as the pressure ratio. Meanwhile, the external losses are made up of disk friction loss, recirculation loss and leakage loss. All the external losses are associated with minor flows leaking from the main flow and assumed as the energy loss outside the main flow path. Thus, it only affects the efficiency of the compressor.

Each loss mechanism of internal loss is as follows: (1) Incidence loss is derived from non-uniform incidence along the leading edge. It increases the relative velocity in the tangential component at the entrance and has the minimum loss at the optimum incidence angle. (2) Blade loading loss is the fluid momentum loss that is associated with blade surface boundary layer growth in decelerating flows. It is also related to flow separation in the impeller. (3) Skin friction loss is similar to enthalpy loss due to the wall friction of turbulent flow in a duct. However, it does not include the effect of blade surface boundary layers. (4) Clearance loss is defined as the loss due to the flow of fluid from the pressure surface of the blade to the suction surface. It is proportional to the ratio of tip clearance to blade height. Tip clearance is the gap between the rotating impeller and the stationary shroud. (5) Mixing loss is also known as wake-jet loss. It is derived from mixing of the high momentum fluid on the pressure side, namely jet, and the low momentum fluid on the suction side, namely wake. (6) Slip loss assumes that the flow cannot be perfectly guided by impeller blades. It causes the flow angle at the impeller

exit leans to the opposite direction of the rotating direction. As a result, the tangential component of absolute velocity at impeller exit decreases as shown in Figure 7.

Each loss mechanism of external loss is as follows: (1) Leakage loss occurs when the leakage flow of tip clearance is re-entered into the blade passage. (2) The recirculation loss is the enthalpy loss brought about the recirculation of low momentum fluid from the vaneless space back into the impeller flow path. (3) Disk friction loss is due to the wall friction between the back surface of the rotating impeller and the stationary surface.

Table 1. List of adopted loss models [24].

Classification of Loss Type		Model
Internal loss	Incidence loss	Conrad
	Blade loading loss	Coppage
	Skin friction loss	Jansen
	Clearance loss	Jansen
	Mixing loss	Johnston and Dean
	Slip loss	Wiesner
External loss	Leakage loss	Aungier
	Recirculation loss	Oh
	Disk friction loss	Nemdili

Figure 5. Internal loss mechanism.

Figure 6. External loss mechanism.

Figure 7. Velocity triangles and flow slip mechanism at impeller exit.

2.3. Loss Model Selection

Loss models can be categorized into internal loss and external loss. The internal loss is defined as how much the actual process is far from the isentropic process. It affects the efficiency as well as the pressure ratio of the compressor. The external loss is defined as the energy loss outside the main flow path. It is associated with minor flows leaking from the main flow through the compressor. It only affects the efficiency of the compressor. Thus, the internal loss models selection is carried out with the pressure ratio performance data and the external loss models selection is based on the efficiency data.

There are three representative loss model sets, which are Aungier loss model set [25], Oh loss model set [26] and Galvas loss model set [27]. However, Aungier is based on pressure loss mechanism while Oh and Galvas are based on enthalpy loss mechanism. Since KAIST-TMD adopted the enthalpy based calculation procedure, Oh set and Galvas set were tested for the loss models. As shown in Figures 8 and 9, the prediction results from KAIST-TMD are compared to the test data of Sandia National Laboratory (SNL) to validate the adopted loss models. According to Figure 8, the Oh set provides good agreement with experimental data on the pressure ratio while the Galvas set is less satisfying. The difference in predictions is due to the skin friction loss calculated from the Galvas model which seems to over-predict the values. Thus, KAIST-TMD adopted the internal model set proposed by Oh.

Figure 8. Predicted pressure ratio results with SNL test data [24].

In Figure 9, the Oh set predicts slightly higher efficiency than the test data while the Galvas set provides poor agreement with the efficiency data. Even though the Oh set is reasonable, a modification was necessary on the disk friction loss model because it assumes a constant disk gap to radius ratio. The original disk friction model in the Oh set is developed by Daily and Nece [28] and the ratio of disk gap to the radius in the disk friction coefficient is not taken as a variable. Thus, the disk friction loss is insensitive to the ratio of disk gap to radius despite the ratio plays a crucial role in the loss mechanism. From the literature survey, an empirical disk friction loss model suggested by Nemdili [29] shows better agreement than that of Daily and Nece as it can be observed from the Figure 9 Oh set with

modification. Table 1 summarizes the modified loss model set from Oh's suggestion and this set is applied to KAIST-TMD code.

Figure 9. Predicted total to static efficiency results with SNL test data [24].

2.4. Selection of Analysis Conditions

Typically, a centrifugal compressor is known to show better efficiency for higher specific speed and larger scale. The specific speed is defined in Equation (9) and uses to estimate an isentropic efficiency, rotational speed and the number of stages of a compressor. The increase in specific speed reduces the flow resistance in an impeller because the flow passage tends to be shorter and wider as specific speed increases. Figure 10 supports this physical insight. Since this study only covers the centrifugal compressor, the specific speed of 0.64, which is the highest in the radial type of Balje's diagram, was chosen [30]. The scale effect can explain through the ratio of tip clearance to blade height. The tip clearance loss and the leakage loss decrease as the system scale increases since the ratio of tip clearance to blade height decreases as the system power capacity increases. Figure 11 represents the best achievable compressor efficiency for different system power class. The minimum tip clearance was selected considering the manufacturing tolerance. Although compressor efficiency increases as the system scale grow, in large systems, an axial type is typically recommended rather than a radial type. Therefore, in this study, a compressor for 10MWe class power conversion system was selected to demonstrate the feasibility of the current study [31].

$$n_s = \frac{NQ^{0.5}}{(gH_{ad})^{0.75}} \quad (9)$$

Blade backsweep angle at impeller exit and static conditions at compressor inlet were chosen as the design parameters for the sensitivity analysis because those are important variables that determine the S-CO₂ compressor outlet conditions. Compared with the conventional compressor, the inlet operating condition is regarded as a very important design factor in the S-CO₂ compressor due to the steep change of thermodynamic properties near the critical point. Figure 12 shows the compressibility factor variation in the S-CO₂ compressor operating range. At CO₂ critical point, compressibility factor has the lowest value of 0.23 and it varies up to 0.65 with increasing temperature and pressure from the critical point. The inlet conditions of the analyzed compressor were selected while considering the compressibility factor. Table 2 summarizes the operating conditions and main design parameters. The design conditions represent the inlet and outlet conditions for the compressor design derived from five different simple recuperated cycle designs. Since the variation of thermodynamic properties near the critical point depends on the compressibility factor, the compressibility factor was used as the representative value for the inlet condition. It is noted that the outlet pressure of all five compressors is

all fixed at 20 MPa when designing five different cycles studied in this paper. In order to minimize the kinetic energy loss at the compressor inlet, the inlet absolute velocity angle was chosen to be 0° . To observe the effect of the backsweep angle change on the compressor performance, the range of 0° to -77° was selected as the analysis range, which encompasses the conventional -50° . Figure 13 is an example of an S-CO₂ compressor impeller after applying the design conditions of case 1 and the design parameters are summarized in Tables 2 and 3 summarizes the results of cycle analysis with recuperated Brayton cycle layout in each case.

Figure 10. Peak total to total efficiency with respect to a specific speed.

Figure 11. Peak total to total efficiency with respect to various power class.

Figure 12. Compressibility factor contour under the S-CO₂ compressor operating range.

Figure 13. Impeller geometry of S-CO₂ compressor generated by KAIST-TMD.

Table 2. Operating conditions and selected design parameters.

Design Conditions					
Case Number	Case 1	Case 2	Case 3	Case 4	Case 5
Working fluid	S-CO ₂	S-CO ₂	S-CO ₂	S-CO ₂	S-CO ₂
Target total pressure [MPa]	20	20	20	20	20
Compressibility factor [-]	0.23	0.31	0.41	0.51	0.60
Inlet static temperature [°C]	31	33	32	36	45
Inlet static pressure [MPa]	7.4	7.7	7.4	7.4	7.4
Design Parameters					
Compressor type	Radial type		Impeller type		Unshrouded
Ratio of inlet hub to shroud	0.2		Tip clearance [mm]		0.15
Number of full blades	14		Specific speed [-]		0.64
Impeller inlet absolute angle	0°		Impeller outlet backsweep angle		0~77°
Ratio of impeller radius to vaneless space radius	1.05	Ratio of impeller blade height to diffuser blade height			1.00

Table 3. Cycle design results of recuperated Brayton cycle.

Cycle Design Results					
Case Number	Case 1	Case 2	Case 3	Case 4	Case 5
Generating power [MWe]	10.0	9.97	10.1	9.95	9.56
Cycle efficiency [%]	33.4	33.2	33.7	33.1	31.9

3. Results

3.1. Effects of Large Backsweep Angle on S-CO₂ Compressor Performance

Compressor performance was examined when the angle varied from 0° to −77° while using the design parameters summarized in Table 2. As shown in Figure 14, the S-CO₂ centrifugal compressor showed the best efficiency at −70°, which is larger than the typical design value for the air centrifugal compressor. Also, in all cases, the total to total efficiency tends to increase as its angle at the impeller exit increases. The S-CO₂ compressor has the best efficiency of 84% at the backsweep angle of −70°. It is an increase of about 3% in comparison with the recommended design region. Despite the rapid change in properties near the critical point, the compressor efficiency is insensitive to the change in the compressibility factor.

Figure 14. Total to total efficiency sensitivity to backsweep angle.

Due to the structural issue of the impeller in the low-density fluids, such as air and helium, the backsweep angle is generally limited to −50°. Figure 15 explains why large backsweep angle involves high centrifugal stress level. The angle of absolute velocity leaving the impeller decreases as the backsweep angle increases when it operates under the same operating conditions. As a result, a higher peripheral speed is required as the backsweep angle to maintain identical outlet pressure and, consequently, it causes greater stresses. Figure 16 summarizes the preliminary structural safety margin using the tip speed which is an important criterion for the centrifugal stress level. In spite that the safety margin decreases as the compressibility factor increases, an S-CO₂ compressor is generally less affected by the structural issues in all cases. The structural limit of centrifugal stress is set to be the same value of the air centrifugal compressor for the same design variables summarized in Table 2

and -50° of backward sweep angle. Also, Figure 17 supports why the S-CO₂ compressor has smaller centrifugal stress than the air compressor thermodynamically. S-CO₂ requires only a few hundredths of the enthalpy of air when it increases the same amount of pressure. It means that the work consumption for compression is significantly reduced when the S-CO₂ compressor increases the same amount of pressure compared to the air compressor. Thus, smaller tip speed is required and this results in smaller centrifugal stress in the S-CO₂ compressor.

The adopted loss models do not cover irreversibility caused by shock wave because it assumes a centrifugal compressor is operating in the subsonic region. Since the speed of sound is substantially decreased near the critical point as shown in Figure 18, a high Mach number may occur in the low-speed region. As aforementioned, the radius of an impeller increases as the backsweep angle at impeller exit increases. Thus, the relative Mach number needs to be checked whether it exceeds unity or not. Figure 19 shows that the relative Mach numbers are under the subsonic region in all cases. It proves that the additional shock loss does not need to be included and the results of Figure 14 are still valid. Also, because of the low Mach number in the S-CO₂ compressor, the variation of compressibility factor does not impact on its aerodynamic design in spite of its operation near the critical point where the compressibility factor change is substantial.

In order to understand the reason why the backsweep angle change affects the efficiency, the compressor loss distribution with varying angle was analyzed as shown in Figures 20–23. It is again noted that the percentage of blade loading loss is minimized at -70° which achieves the highest efficiency. Figure 20 shows the change in blade loading loss, clearance loss and mixing loss with increasing backsweep angle. All three losses are associated with the pressure loading on the blade. The blade loading loss is due to the flow separation resulting from the pressure load difference between the suction side and the pressure side. In this respect, the inclination of the blades to the radial direction reduces the pressure loading and it alleviates the secondary flow and the tip clearance flow. As the backsweep angle increases, less pressure loading per unit length on the blade is expected, which results in reducing the strength of the vortex at the blade tip. This means that the clearance loss decreases as the backsweep angle increases. Mixing loss is derived from mixing of the high momentum fluid on the pressure side and the low momentum fluid on the suction side. Decrease of pressure load means a reduction in the difference of fluid momentum between the pressure side and suction side, and the mixing loss decreases as the backsweep angle increases. This physical insight corresponds to the results of Figure 20. Also, as the blade angle becomes more-flat, the radius increases as shown in Figure 24. This is because the pressure ratio of the compressor is assumed to be constant even if the backsweep angle changes. As the backsweep angle increases, smaller tangential component of absolute velocity is obtained. As a result, the diameter of the impeller is increased, as shown in Figure 24, to maintain the same outlet pressure. Thus, as shown in Figures 21 and 22, the skin friction loss, leakage loss, and the disk friction loss tend to increase because those are proportional to the impeller size. The compressor design is optimized to have the best efficiency as the backsweep angle changes. Therefore, the optimal incidence angle was chosen to minimize incidence loss, and consequently, as shown in Figure 23, the incidence loss is always zero regardless of the change in the backsweep angle.

Figure 25 shows the effects of the off-design performance and the surge limit while backsweep angle varies. As shown in Figure 15, if the mass flow rate decreases, the angle of relative velocity, W_2 , remains the same. On the other hand, the tangential component of the absolute velocity, C_{w2} , increases. Since the product of U_2 and C_{w2} has increased in comparison with the high mass flow rate condition, the work per unit of mass flow rate at the low mass flow rate condition has increased considerably. Thus, a backward sweep impeller can achieve higher pressure ratio as the mass flow rate is decreased. Also, this feature allows stable compressor operation in wider range because the surge limit is conservatively set as the point where the gradient of pressure ratio becomes zero.

Figure 15. Effect of the backsweep angle at impeller exit on velocity triangles.

Figure 16. Distribution of tip speed at impeller exit.

Figure 17. dh/dP of S-CO₂ and air with respect to temperature.

Figure 18. Sound speed variation near the critical point.

Figure 19. Distribution of relative Mach number.

Figure 20. Distribution of internal losses with backsweep angle variation.

Figure 21. Distribution of internal losses with backsweep angle variation.

Figure 22. Distribution of external losses with backsweep angle variation.

Figure 23. Percentage of cumulative loss depending on backsweep angle variation.

Figure 24. Impeller radius change with backsweep angle variation.

Figure 25. Effects of off-design performance and surge margin with backsweep angle variation.

3.2. Impact of Large Backsweep Angle S-CO₂ Compressor on Cycle Performance

Figure 26 shows the improvement in cycle efficiency using higher backsweep angle compressor depending on CIT and TIT variations. The cycle optimization was performed with KAIST-ESCA code described in Ref. [32]. The optimization process is based on the adjoint method while solving all the general thermodynamic relations in a cycle [32]. Table 4 summarizes the assumed conditions for the cycle analysis. The pressure ratio was limited to 2.7 to avoid the system minimum pressure from falling below the critical point. Since the dry cooling system is mainly installed in desert or arid areas, the compressor inlet temperature can be more than 50 °C [3]. The compressor efficiency is selected from the analysis results when the backsweep angle is 50° and 70° in the previous section.

As it was expected, the impact of compressor efficiency on the cycle efficiency was greater in the dry cooling case as CIT increases. Again, this is because S-CO₂ shows a steep rise in density near the critical point and this increase reduces the compressor work significantly. Thus, the improvement of compressor efficiency has relatively minor effect on the cycle performance if compression process occurs near the critical point because the compressor work is small, to begin with. In contrast, the compressor work increases when CIT is higher because the density decreases as the operation point moves away from the critical point. This means that the improvement of compressor efficiency is a more attractive option in dry cooling than in water cooling case. Thus, a large backsweep angle impeller becomes an essential design option for the S-CO₂ Brayton cycle when the compressor inlet temperature is expected to be further away from the critical point.

Figure 26. Contour of cycle efficiency increase.

Table 4. Design conditions for cycle optimization.

Cycle Design Constraints			
Maximum Pressure [MPa]	20	η_{TB} [%]	90
TIT [°C]	350~750	ε of recuperators [%]	95
Pressure drop in heat exchangers [MPa]	0.15	Generator efficiency [%]	98
η_{COM} [%]	81/84	CIT [°C]	31~70

4. Conclusions

The desire to minimize water consumption led to the power plant integrated with a dry cooling system. This is also true for a power plant using supercritical CO₂ power cycle. However, this will result in a higher compressor inlet temperature (CIT). Thus, an S-CO₂ Brayton cycle with the dry cooling system inevitably faces substantial deterioration of the cycle efficiency due to losing the benefit of reduced compression work when dry cooling is necessary. This study suggests how to improve aerodynamic performance of an S-CO₂ centrifugal compressor when the compressor inlet temperature can be high to recover the cycle efficiency eventually.

A conventional air centrifugal compressor can achieve higher efficiency as backsweep angle increases, however, the maximum allowable backsweep angle is restricted due to the structural issue to the values of about -50° . In this study, an S-CO₂ centrifugal compressor performance was examined while changing the backward sweep angle at impeller exit to investigate the same maximum backsweep angle should be applied to the S-CO₂ centrifugal compressor as in the air centrifugal compressor. It was found that an S-CO₂ compressor can achieve the highest efficiency at -70° backsweep angle, which is greater than the typical design value of the air centrifugal compressor. This is because an S-CO₂ compressor can be less limited by the structural integrity compared to the air compressor because S-CO₂ requires only a few hundredths of the enthalpy rise of air when it increases the same amount of pressure. Consequently, the S-CO₂ compressor has lower relative Mach number even though S-CO₂ has lower speed of sound near the critical point. It was also quite surprising that the variation of compressibility factor near the critical point does not impact on its total to total efficiency since its Mach number is still lower than unity. A backward sweep impeller can achieve higher pressure ratio and operate stably in wider range as the mass flow rate is decreased, thus it is recommended to have larger backsweep angle when designing an S-CO₂ centrifugal compressor.

As further works, the suggested concept will be proved by more detail structural analysis and the compressor performance test experimentally.

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Nomenclature

C	Velocity
C_f	Friction coefficient
CP	Compressor
H_{ad}	Adiabatic head
HTR	High temperature recuperator
IHX	Intermediate heat exchanger
LTR	Low temperature recuperator
M	Mach number
MC	Main compressor
N	Rotational speed (rad/s)
P	Pressure
P_c	Critical pressure
PC	Pre-cooler
Q	Volumetric flow rate (m ³ /s)
RC	Recompressor
RCP	Recuperator
T	Temperature
TB	Turbine
T_c	Critical temperature
U	Tip speed or peripheral speed
V	Velocity
b	Ratio of vaneless diffuser inlet width to impeller exit width
β_2	Backsweep angle at impeller exit
g	Gravitational acceleration
h	Enthalpy
n_s	Specific speed
r	Radius
α	Absolute angle to meridional direction
β	Relative angle to meridional direction
γ	Ratio of specific heats
ρ	Density
ϕ	Inclination of meridional streamline to machine axis

Subscript

i	Ideal
m	Meridional component
o	Stagnation
s	Static
w	Tangential component
0	Zero backsweep angle
1	Impeller inlet
2	Impeller outlet

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