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PRELIMINARY EXPERIMENTAL STUDY OF CO_2 BASED MIXTURE SUPERCRITICAL POWER CYCLE

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ABSTRACT

The supercritical carbon dioxide (sCO_2) Brayton power cycle has been receiving worldwide attention due to the high thermal efficiency and compact system configuration. Because of the incompressible liquid like characteristic (e.g. high density, low compressibility) of the CO_2 near the critical point (30.98℃, 7.38MPa), sCO_2 Brayton cycle can achieve high efficiency by reducing compression work. The system is sufficiently advanced, but has drawbacks in terms of minimum temperature and high operating pressure. These disadvantages can be improved by mixing working fluid with different materials. In this study, the authors evaluated the performance improvement by changing critical point and operating conditions. The higher molecular weight substances (SF_6, Organic refrigerants, Hydrocarbons…) can be utilized to increase the mixture critical temperature and lower the mixture critical pressure. In this study, thermodynamic evaluations of mixing CO_2 with difluoromethane (R-32) or toluene are considered. Also the preliminary experimental verification of the CO_2 + R-32 case is conducted to show the technical feasibility of the suggested ideas.

INTRODUCTION

In recent years, interests in research on high-efficiency energy conversion technology have increased substantially to resolve global warming problems and respond to the increase in energy demand. The supercritical carbon dioxide Brayton power cycle technology has been thought to be one of the promising technical solutions to these issues [1- 3].

The sCO_2 power conversion cycle can achieve high efficiency by reducing compression work due to the high density and low compressibility characteristics near the critical point (30.98℃, 7.38MPa). Moreover, the power cycle can be designed in compact size due to the high density of the working fluid. Moreover, the compact sCO_2 power system can be utilized as modularized distributed power generation or advanced marine propulsion technology.

However, the sCO_2 power cycle has intrinsic limitation on the minimum temperature which is at the critical temperature of the CO_2. Due to the small temperature difference between ultimate heat sink and system minimum temperature, the sCO_2 power system requires a large amount of cooling flow rate or massive heat exchanger to dissipate the waste heat especially via air-cooling in hot regions.

In order to improve the sCO_2 system regarding the minimum temperature limitation, a research on the CO_2 based gas mixture power cycle has been conducted previously. Jeong et al. mixed CO_2 with N_2, O_2, He or Ar gas to evaluate the performance of the binary mixture gas supercritical Brayton cycles [4]. Yin et al. suggested CO_2-SF_6 mixture for geothermal power plants [6-7]. Hu et al. and Wu et al. analyzed CO_2-based binary mixture with organic fluids (e.g., butane, cyclohexane, ethyl fluoride) [8-9]. Also the authors presented the thermodynamic analysis of CO_2 mixture with substances such as SF_6, R-123, R-134a, R-22, R-32 and toluene power cycle in supercritical state to improve the system efficiency and reduce the difficulty of air-cooled waste heat removal method [14].

In this study, thermodynamic evaluations of mixing CO_2 with difluoromethane (R-32) or toluene are considered to configure trans-critical power cycles at 150℃ and 300℃ of maximum temperature, respectively. Also the preliminary experimental verification of the CO_2 + R-32 case (12% of mass fraction) is conducted to show the technical feasibility of the suggested ideas.

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CRITICAL POINT OF VARIOUS SUBSTANCES

According to the thermodynamic definition, the critical point is a temperature and pressure of a material beyond which there is no longer any difference between the liquid and gas phases. Pure substances have their unique critical point and it is known that the critical point can be changed by mixing with other substances.

The most basic mathematical model for estimating the mixture critical point is the W. B. Kay's equilibrium critical point model [10]. It is a mole fraction based linear interpolation method as shown in the following equations.

$$(P_c)_{\text{mix}} = \sum_i y_i P_{ci} \tag{1}$$

$$(T_c)_{\text{mix}} = \sum_i y_i T_{ci} \tag{2}$$

Where, y_i is mole fraction of the i^{th} component

It can be utilized for gas dynamics problem for ideal case but the real gas equilibrium critical point locus shows different locus compared to the suggested linear interpolation methods [11]. Furthermore, it is difficult to identify generalized formulation of critical locus because of the actual phenomenon differs depending on the substances and their combinations. There exist six different principal types, but the basic type of binary mixture critical locus is shown in the following figure. In Figure 1, the saturation line and critical point of each substance are marked as solid line with the empty circle. The dotted line shows the locus of mixture critical point depending on the mixing ratio of the two substances. At the certain mixing ratio, the two-phase region is shown through the bubble and dew lines in temperature-pressure (T-P) graph rather than a single curve.

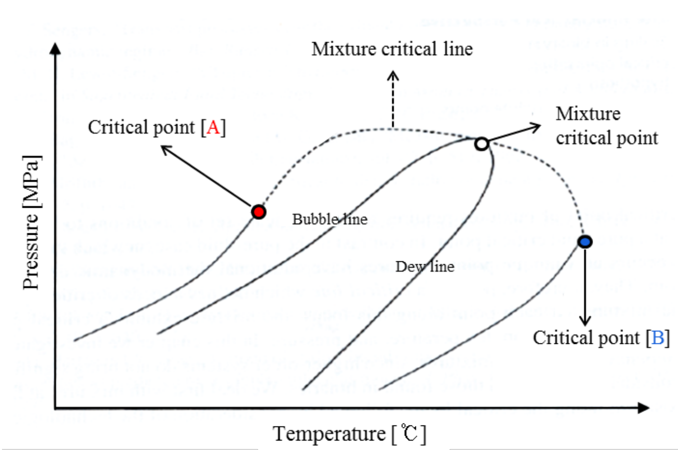

Figure 1: An example of critical locus of binary gas mixture [11]

In this study, the thermodynamic analysis is carried out based on the most reliable NIST REFPROP (ver. 10.0) property data base. The NIST data base is an international standard in thermal fluid engineering which is based on extensive experimental results [12, 13]. As examples of critical point change of CO_2 based mixtures, the critical loci are plotted in T-P graph while varying the mixing fraction.

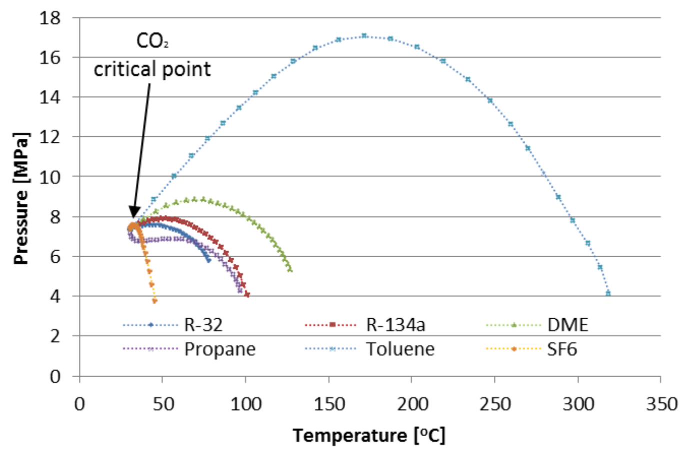

Figure 2: Critical locus of CO_2 based mixtures

Among the pure fluids available in NIST data base, the authors defined isentropic enthalpy difference index ($\left.\frac{\partial h}{\partial P}\right|_s$) and compared the values at compressor and turbine inlet conditions for the preliminary screening criteria on choosing substances [14]. The compressor and turbine inlet conditions are 7.41MPa, 31℃ and 20MPa, 300℃ respectively. Also the compressor inlet density is compared together to minimize the compression work at the compressor. The information of critical point, density and isentropic enthalpy difference index are tabulated in Annex A.

As shown in Annex A, high density and low isentropic enthalpy difference at compressor, high isentropic enthalpy difference at turbine, high critical temperature substances are highlighted in bold letters. Those substances were rearranged with basic information in Table 1.

Organic refrigerants, including R-32 and R-134a, have a critical temperature range of 50-200℃ and a critical pressure range of 3-5 MPa. Due to the suitable range for engineering purpose, those refrigerants are mainly used as a working fluid for refrigeration system or heat pump applications.

Dimethyl ether and propane are sometimes regarded as refrigerants, but they are mainly used as fuels or propellants due to the high flammability [15-16]. Toluene is an organic compound used primarily as a solvent, but recently it has been recommended as a suitable working fluid for high temperature organic Rankine cycle (ORC) applications. [17] The SF_6 is heavy gas which is mainly used in high voltage circuit breakers due to the small electric conductivity and stability in high temperature [18].

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Table 1: Information of each substance

Substance	Molar mass [kg/kmol]	T_c [℃]	P_c [MPa]	D_c [kg/m^3]
CO_2	44.01	30.98	7.3773	467.6
R-32 (CH_2F_2)	52.02	78.11	5.7820	424.0
R-134a (CF_3CH_2F)	102.03	101.06	4.0593	511.9
Dimethyl ether (CH_3OCH_3)	46.07	127.23	5.3368	273.7
Propane ($CH_3CH_2CH_3$)	44.10	96.74	4.2512	220.5
Toluene ($C_6H_5CH_3$)	92.14	318.60	4.1263	292.0
SF_6	146.06	45.57	3.7550	742.3

MIXTURE PROPERTIES

In this study, thermodynamic studies of CO_2 power cycle mixing with R-32 or toluene are first considered. The preliminary screening results based on the existing mixture properties available in open literature are shown.

As mentioned above, estimation of the critical locus or thermodynamic properties of mixture is difficult to generalize. Microscopically, it is due to the differences of the physical characteristics from the molecular structure and intermolecular interactions between mixtures. However, the microscopic approach on the thermodynamic system such as establishing the equation of state or mixing rule is beyond the scope of the study. Therefore, the calculated mixture property results of the REFPROP software were verified by comparing with the existing experimental results.

Previously, the experimental studies on vapor-liquid equilibria data and critical locus of CO_2+R-32 and CO_2+Toluene cases were conducted by many chemical engineers [19-21]. Those studies are conducted in purpose of using mixtures for refrigeration system or extraction process by modifying their thermodynamic characteristics.

The authors aimed to increase the critical temperature of CO_2 by mixing additives to enable condensation at 31℃, it is important to predict accurately the bubble and dew pressure. Comparisons of REFPROP calculated data with experimental data were plotted. As shown in the figures, the mixture of CO_2+R32 showed reliable estimation on bubble and dew pressure. Also the critical locus showed reasonable agreement. On the other hand, the estimation of CO_2+Toluene mixture case showed unreliable results compare to the experimental data at high temperature regions and when CO_2 fraction is small (CO_2 mass fraction lower than 0.7). However, the estimation of critical locus with REFPROP was reliable when it is compared to the two experimental data sets.

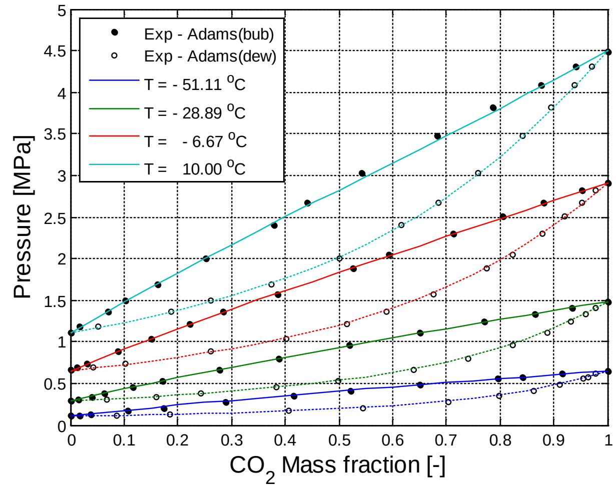

Figure 3: Comparison of calculated bubble pressure and dew pressure of CO_2+R-32 mixture from REFPROP with experimental data [19]

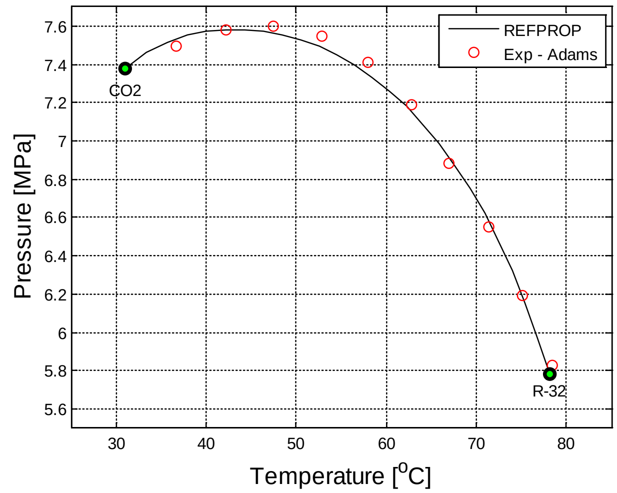

Figure 4: Comparison of calculated critical locus of CO_2+R-32 mixture from REFPROP with experimental data [19]

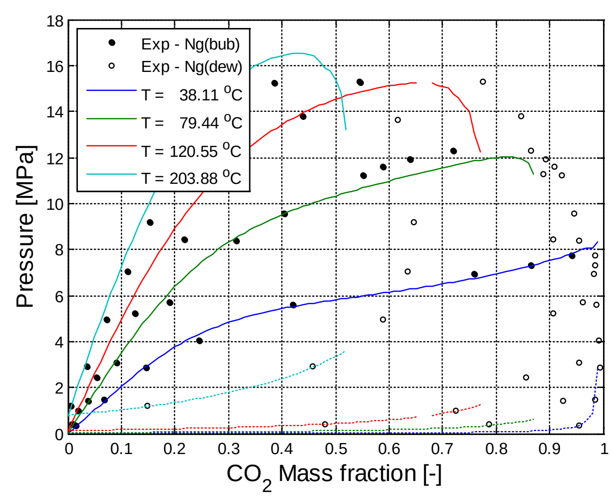

Figure 5: Comparison of calculated bubble pressure and dew pressure of CO_2+Toluene mixture from REFPROP with experimental data [20]

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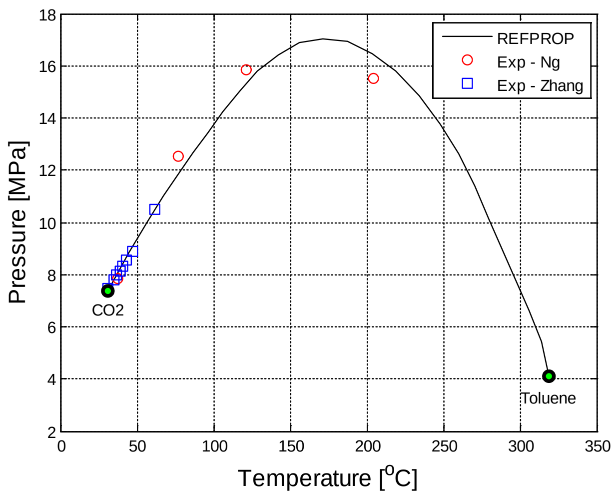

Figure 6: Comparison of calculated critical locus of CO_2+Toluene mixture from REFPROP with experimental data [21]

Through these verifications, the authors confirmed that the obtained data from the REFPROP package is reliable for CO_2+R-32 case and CO_2+Toluene at CO_2 rich region. In these regions, thermodynamic analysis of binary CO_2 mixture system is conducted.

THERMODYNAMIC ANALYSIS

In order to design and analyze the sCO_2 power cycle, KAIST research team developed an in-house code; namely KAIST_CCD. The developed code has been validated and verified through the previous studies [22, 23]. By utilizing the developed code, thermodynamic analyses of CO_2 based mixture are conducted and compared to the pure CO_2 system's performance.

To compare two thermodynamic systems in fair manner, the cycle was calculated while the lowest temperature was kept at 30℃ and the efficiency of each component was set identical as summarized in the table below. The efficiencies of components were selected as conservative values and the pressure drop was neglected in this study for simplification. Since the pressure losses can't be neglected in an actual system, the authors plan to consider it in the future study.

Table 2: Description of analysis

System type	Trans-critical cycle / Recuperated Trans-critical cycle
Working Fluid	CO_2 / CO_2+R-32 / CO_2+Toluene
Thermal input	1 MW_{th}
Mass flow rate	4-5 kg/s
Maximum Pressure	20.0 MPa
Maximum Temperature	150 ℃ / 300 ℃
Minimum Temperature	30 ℃
Compressor Efficiency	80 %
Turbine Efficiency	90 %
Recuperator effectiveness	90 %
Pressure Losses	0 % (Neglected)

The maximum temperatures of the system with addition of R-32 and toluene are selected 150℃ and 300℃ to ensure the thermal stability. In the CO_2 system mixing with toluene case, recuperator was needed to utilize the high turbine outlet temperature. The schematic diagram of each type is shown with T-s diagram in the following figures.

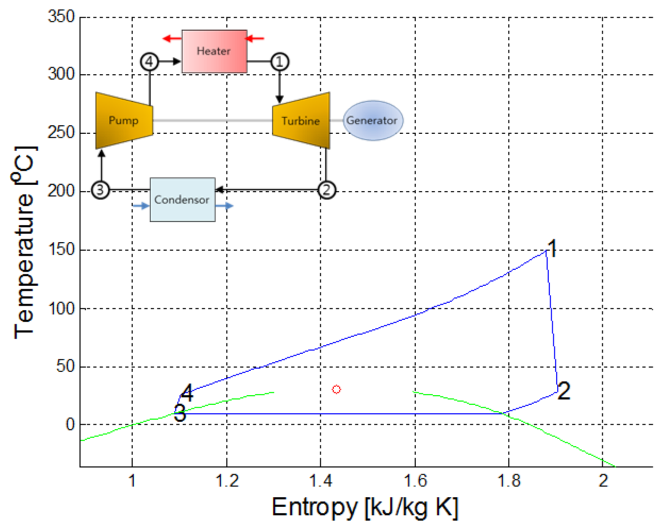

Figure 7: Schematic diagram and T-s diagram of simple trans-critical cycle

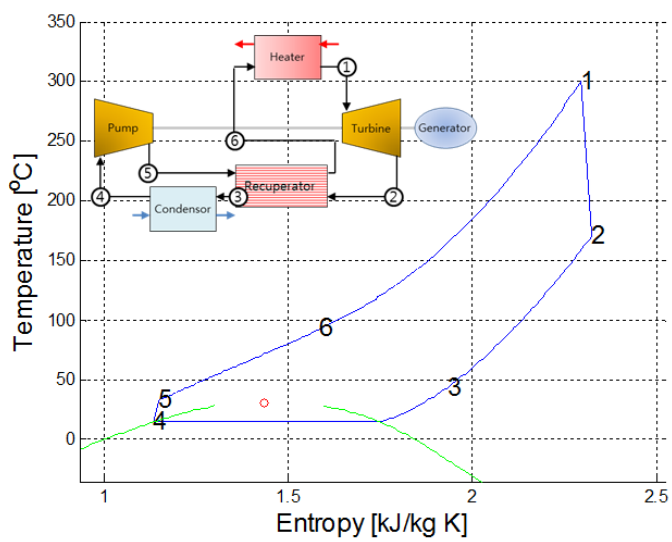

Figure 8: Schematic diagram and T-s diagram of recuperated trans-critical cycle

From thermodynamic calculations, variations of system efficiency with respect to different additive compositions are

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plotted in the following figures. The system efficiency is 1st law thermal efficiency which is expressed by equation (3).

$$\text{Thermal efficiency } (\eta_{th}) = \frac{W_{net,out}}{Q_{in}} \quad (3)$$

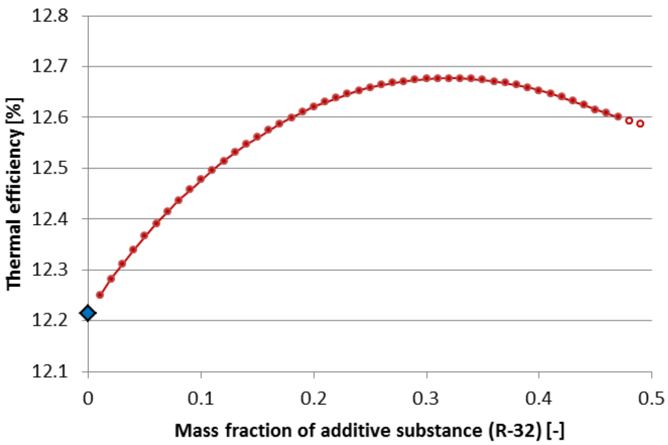

Figure 9: Efficiency change of CO_2+R-32 trans-critical cycle with respect to R-32 mass fraction

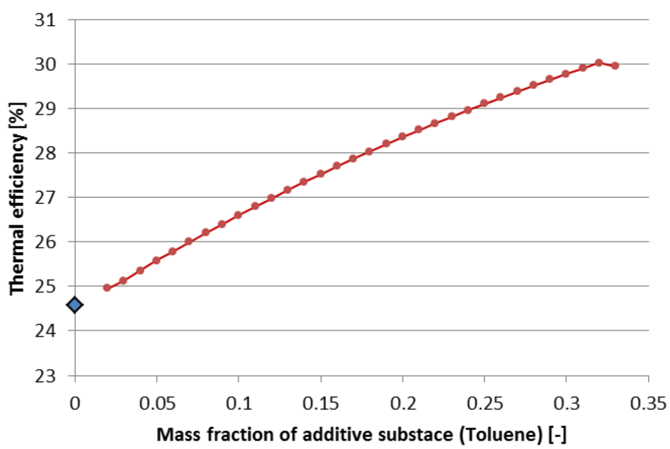

Figure 10: Efficiency change of CO_2+Toluene recuperated trans-critical cycle with respect to the toluene mass fraction

When R-32 is mixed with CO_2 and the maximum temperature is 150℃, the calculation showed an increase in efficiency over the pure CO_2 system. It was found that mixing up to 32% can increase the efficiency by 0.46% point. The results of adding R-32 more than 48% are excluded because the droplets may occur at the turbine outlet condition.

For the maximum temperature of 300℃ and when toluene is mixed, the calculation result also showed an increased efficiency compared with the pure CO_2 system. The calculated system efficiency at the maximum mixing ratio (32%) is 30.02%, which is 5.2%p higher than the reference pure CO_2 system. Adding toluene more than 44% are not recommended since the turbine outlet condition may exceed the dew point.

From the thermodynamic studies based on the REFPROP property database, R-32 and toluene showed improvement in the system efficiency compared to the pure CO_2 power cycle. In order to clarify the property uncertainties and confirm the feasibility of the suggested system, experimental tests are conducted with KAIST-S-CO_2PE facility.

EXPERIMENTAL STUDY

The authors conducted a compressor performance test with CO_2+R-32 (0.88:0.12 mass fractions) mixture by utilizing KAIST-S-CO_2PE facility. The experimental facility, which is shown in the following figure, is a sCO_2 power cycle demonstration facility. The test facility was configured to have a simple Brayton cycle layout [23] and it has been utilized for performance test of sCO_2 power cycle components such as centrifugal compressor, PCHE (Printed Circuit Heat Exchanger) and STHE (Shell and Tube Heat Exchanger) types of pre-cooler. The 26kW powered radial compressor was utilized to circulate and compress the working fluid. For the compressor performance test, 234mm diameter shrouded type impeller was operated at constant speed of 3600rpm.

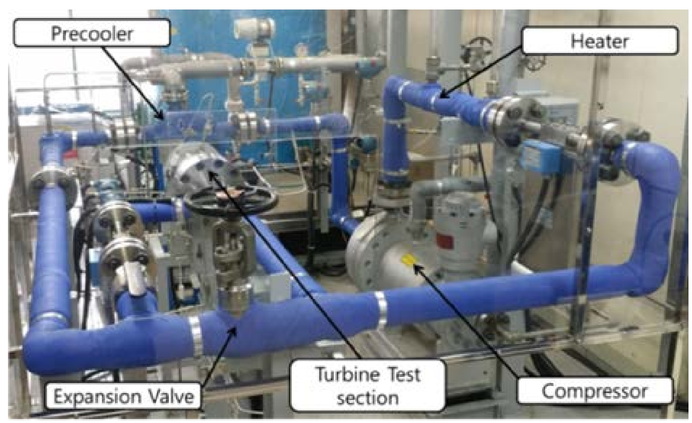

Figure 11: sCO_2 power cycle demonstration facility (KASIT S-CO_2PE facility)

From the previous study [24], pure CO_2 compressor performance test results were accumulated and used as the reference case. 16 different reference test cases are summarized with fluid property in Annex B.

In order to set the mixing ratio of the working fluid accurately, a highly accurate scale (CAS HB-150, Max 150kg, ±10g) was utilized to measure the charged weight of each fluid. Compressor test cases 4, 5, 7 and 10 were selected and re-evaluated using CO_2+R-32 (0.88:0.12 mass fractions) fluid. The largest enthalpy change in the experimental facility occurred in the compressor. Therefore, the authors checked the enthalpy rise by measuring the compressor inlet and outlet pressure and temperature to observe any inconsistencies in the property database.

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EXPERIMENTAL RESULTS

As shown in Table 3, the CO_2+R-32 cases showed 1~2% higher pressure ratio than the pure CO_2 cases. For the pressure measurement, ±0.05% accuracy of pressure transmitter (Rosemount, 3051S1CG5A2) was utilized. Throughout the test range, maximum pressure measurement error is 4.7kPa and uncertainty of pressure ratio is ±0.2%. 1~2% of pressure ratio increase was measured. It seems small increase but it is still meaningful value since it is consistently outside the measurement uncertainty.

Table 3: Experimental results of CO_2 and CO_2+R-32 case

#	T [℃]	P [MPa]	CO_2		CO_2 + R-32 (0.88:0.12)		$\frac{PR_2}{PR_1}$
			Mass flow rate [kg/s]	**PR_1 Pressure ratio** [-]	**Mass flow rate** [kg/s]	**PR_2 Pressure ratio** [-]	
4-1	31	8.4	3.96	1.11	4.15	1.12	1.007
4-2			2.97	1.12	2.96	1.12	1.007
4-3			2.00	1.11	1.93	1.12	1.005
5-1	29.2	7.7	3.94	1.12	3.91	1.13	1.010
5-2			2.99	1.12	2.99	1.14	1.009
5-3			2.02	1.12	2.02	1.13	1.008
7-1	37	8.4	3.88	1.11	3.82	1.12	1.008
7-2			3.16	1.08	3.09	1.11	1.021
7-3			2.29	1.08	2.27	1.11	1.020
7-4			1.61	1.08	1.57	1.1	1.020
7-5			0.82	1.08	0.80	1.1	1.018
10-1	33	7.4	2.81	1.11	2.81	1.12	1.006
10-2			2.09	1.11	2.02	1.12	1.005
10-3			1.35	1.11	1.35	1.11	1.005
10-4			0.65	1.10	0.64	1.11	1.004

To clarify the increase of the pressure ratio, the pressure ratio and pressure difference of the compressor inlet and outlet are plotted in figures 12 and 13.

The red markers represent the test cases of the mixed fluid (CO_2+R-32, 0.88:0.12 mass fractions), and it can be confirmed that the result has a similar tendency to the result of the same high density case of previous pure CO_2 results. The authors concluded that the 1~2% increase of pressure ratio was caused by 10~40% increase of compressor inlet density.

The amount of the performance increase seemed relatively small due to the limitation of low speed compressor, but the difference can be larger in high performance, high speed compressor. Conversely, less work or less speed may be required to raise the same pressure if the CO_2 power system is designed with the mixture.

Moreover, mixing additive will allow CO_2 to liquefy even at temperatures above 31℃. It is expected that the CO_2 based mixture cycle system not only improves system efficiency but also can get around with mechanical issues due to using high-speed compressors for the Brayton cycle by simply replacing compressors with pumps.

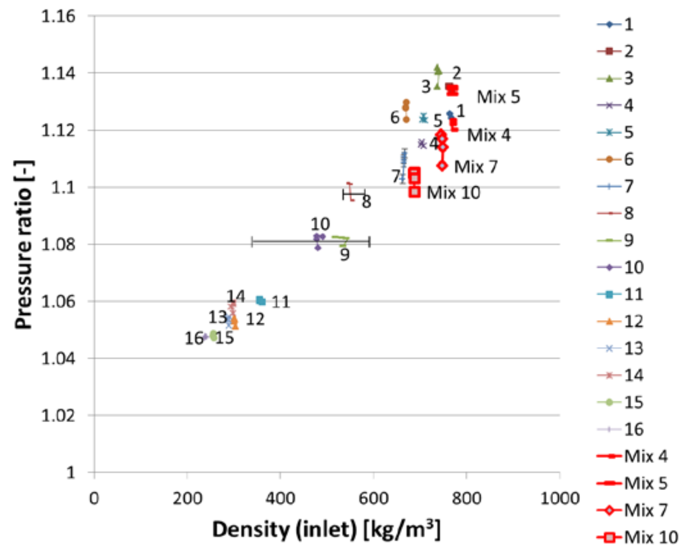

Figure 12: S-CO_2PE compressor performance test result (pressure ratio = P_{out}/P_{in}) in various inlet density conditions

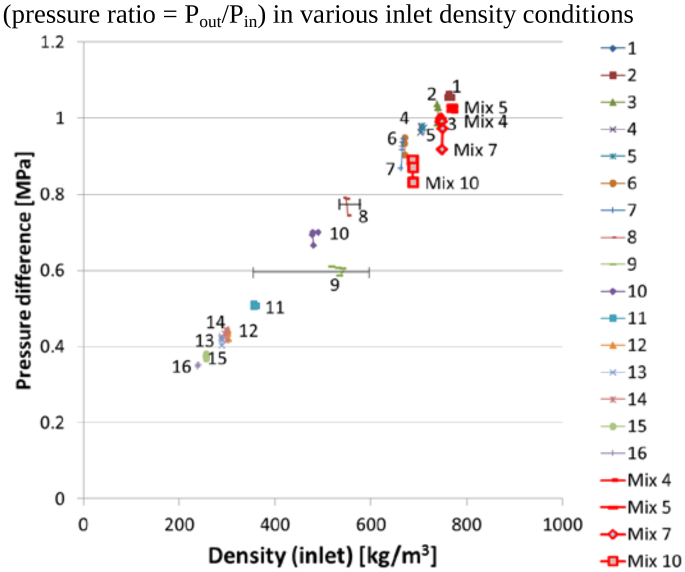

Figure 13: S-CO_2PE compressor performance test result (pressure difference = P_{out}-P_{in}) in various inlet density conditions

SUMMARY AND FUTURE WORKS

In order to improve the thermodynamic performance of the sCO_2 power cycle and reduce the limitation of the cycle minimum temperature, a study on the CO_2 based mixture is conducted. To increase the critical temperature, mixing higher molecular weight substances (such as R-32, R-134a, Toluene, Dimethyl ether…) is investigated. In this study, thermodynamic analyses of CO_2 system by adding R-32 and toluene are conducted. To ensure the thermal stability of additive substances, the maximum temperature of the system was restricted to 150℃, 300 ℃ respectively. At each maximum temperature condition, R-32 case showed 0.46%p increase in thermal efficiency and toluene case showed 5.2%p increase in efficiency compared to the pure CO_2 system.

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Also the experimental test of R-32 case is conducted to show the technical feasibility of the suggested CO_2 based binary mixture power cycle. As expected, the compressor pressure ratio of mixture cases are increased while maintaining the same operating temperature, pressure and mass flowrate conditions to the pure CO_2 cases. The 1~2% increase of pressure ratio was caused by 10~40% increase of compressor inlet density. And it was confirmed that the result has a similar tendency to the results for the high density pure CO_2 cases.

Through a simple compressor performance test, uncertainties of the mixture property were indirectly evaluated near the mixture critical point. The authors concluded that the binary mixtures such as CO_2+R-32 and CO_2+toluene can potentially improve the cycle efficiency by moving critical point of the pure CO_2 system. Also it is expected that, the lowering minimum temperature can further increase the efficiency due to the liquid state compression and maximizing pressure ratio. However, it was noted that the optimum additive substances or composition of addition can be varied depending on the minimum temperature or operating conditions. The detailed study on designing binary mixture system will be conducted next.

Further studies will be carried out on the reduction of exergy destruction with temperature gliding in binary mixture condensation and improvement of cycle operation by utilizing equilibrium conditions. Furthermore, the economic analysis of binary mixture system application will be followed.

NOMENCLATURE

CO_2 : Carbon dioxide
N_2 : Nitrogen
O_2 : Oxygen
DME : Dimethyl ether
SF_6 : Sulfur hexafluoride
R-### : Refrigerant
ORC : Organic Rankine Cycle
NIST : National Institute of Standards and Technology
T_c [℃] : Critical temperature
P_c [MPa] : Critical pressure
D_c [kg/m^3] : Density at critical point
$\left.\frac{\partial h}{\partial P}\right|_s$ [kJ/kg MPa] : Isentropic enthalpy difference index
η_{th} [%] : The 1st law thermal efficiency

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ANNEX A

Substance	**Molar mass** [kg/kmol]	$\mathbf{T_c}$ [℃]	$\mathbf{P_c}$ [MPa]	$\mathbf{D_c}$ [kg/m^3]	**A Compressor inlet density** [kg/m^3]	**B Compressor inlet** $\left.\frac{\partial h}{\partial P}\right\|_s$ [kJ/kg MPa]	**C Turbine inlet** $\left.\frac{\partial h}{\partial P}\right\|_s$ [kJ/kg MPa]	$\mathbf{A \div B \times C}$ [kg/m^3]
CO_2	44.0	31.0	7.377	467.6	573.2	1.74	5.21	1711.9
R-11	137.4	198.0	4.408	554.0	1481.6	0.675	1.191	2613.7
R-12	120.9	112.0	4.136	565.0	1328.6	0.753	1.662	2933.4
R-13	104.5	28.9	3.879	582.9	1003.5	0.997	2.296	2312.2
R-14	88.0	-45.6	3.750	625.7	335.3	2.983	2.976	334.5
R-21	102.9	178.3	5.181	526.0	1372.4	0.729	1.530	2881.9
R-22	86.5	96.1	4.990	523.8	1203.9	0.831	2.358	3417.3
R-23	70.0	26.1	4.832	526.5	844.1	1.185	3.316	2362.6
R-32	**52.0**	**78.1**	**5.782**	**424.0**	**970.4**	**1.031**	**3.949**	**3718.4**
R-41	34.0	44.1	5.897	316.5	588.8	1.698	6.295	2182.8
R-113	187.4	214.1	3.392	560.0	1569.1	0.637	1.006	2478.0
R-114	170.9	145.7	3.257	580.0	1468.0	0.681	1.240	2672.1
R-115	154.5	80.0	3.129	614.8	1325.7	0.754	1.547	2718.3
R-116	138.0	19.9	3.048	613.3	1047.3	0.955	1.897	2081.0
R-123	152.9	183.7	3.662	550.0	1469.6	0.680	1.203	2597.6
R-124	136.5	122.3	3.624	560.0	1370.3	0.730	1.542	2895.4
R-125	120.0	66.0	3.618	573.6	1230.5	0.813	1.963	2972.8
R-134a	**102.0**	**101.1**	**4.059**	**511.9**	**1222.0**	**0.818**	**2.071**	**3092.8**
R-141b	116.9	204.4	4.212	458.6	1237.5	0.808	1.398	2140.7
R-142b	100.5	137.1	4.055	446.0	1120.5	0.892	1.877	2356.4
R-143a	84.0	72.7	3.761	431.0	959.4	1.042	2.587	2382.0
R-152a	66.1	113.3	4.517	368.0	908.5	1.101	2.911	2402.6
R-218	188.0	71.9	2.640	628.0	1379.8	0.725	1.404	2673.1
R-227ea	170.0	101.8	2.925	594.3	1413.6	0.707	1.417	2830.9
R-236fa	152.0	124.9	3.200	551.3	1374.0	0.728	1.457	2750.2
R-245fa	134.0	154.0	3.651	516.1	1344.7	0.744	1.491	2695.8
R-365mfc	148.1	186.9	3.266	473.8	1262.3	0.792	1.371	2185.3
R-1216	150.0	85.8	3.150	583.4	1345.5	0.743	1.613	2919.7
R-1233zd	130.5	165.6	3.573	478.9	1269.7	0.788	1.481	2387.6
R-1234yf	114.0	94.7	3.382	475.6	1115.0	0.897	1.956	2431.3
R-1234ze	114.0	109.4	3.635	489.2	1177.6	0.849	1.912	2651.5
RC-318	200.0	115.2	2.778	620.0	1521.3	0.657	1.240	2869.0
RE-245cb2	150.0	133.7	2.886	499.5	1281.7	0.780	1.469	2413.9
RE-245fa2	150.0	171.7	3.433	515.0	1390.5	0.719	1.339	2588.8
RE-347mcc	200.1	164.6	2.476	524.1	1416.4	0.706	1.181	2369.1
DME	**46.1**	**127.2**	**5.337**	**273.6**	**665.4**	**1.503**	**3.973**	**1759.2**
Propane	**44.1**	**96.7**	**4.251**	**220.5**	**501.3**	**1.995**	**4.668**	**1173.1**
Benzene	78.1	288.9	4.907	304.7	873.5	1.145	1.682	1283.7
Toluene	**92.1**	**318.6**	**4.126**	**292.0**	**862.5**	**1.159**	**1.608**	**1196.4**
MM	162.4	245.5	1.939	304.4	766.4	1.305	1.738	1021.0
MDM	236.5	290.9	1.415	256.7	823.0	1.215	1.603	1085.5
SF_6	**146.1**	**45.6**	**3.755**	**742.3**	**1432.6**	**0.698**	**1.650**	**3386.5**

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ANNEX B

Table B. S-CO_2PE pure CO_2 test cases and inlet fluid property

Test case	**Temperature** [℃]	**Pressure** [MPa]	**Density** [kg/m^3]	**Speed of sound** [m/s]	**Comp. Factor** [-]
1	27.0	8.44	765	361	0.19
2	25.6	7.84	765	355	0.18
3	26.0	7.30	739	320	0.17
4	31.1	8.44	705	302	0.21
5	29.2	7.84	708	297	0.19
6	29.0	7.30	671	253	0.19
7	33.0	8.44	665	269	0.22
8	33.0	7.80	548	189	0.25
9	31.0	7.38	533	144	0.24
10	37.0	8.45	482	193	0.30
11	39.3	8.44	358	197	0.40
12	40.1	8.20	302	203	0.46
13	37.0	7.80	289	199	0.46
14	33.0	7.45	299	191	0.43
15	40.1	7.80	257	208	0.51
16	37.3	7.39	239	208	0.53

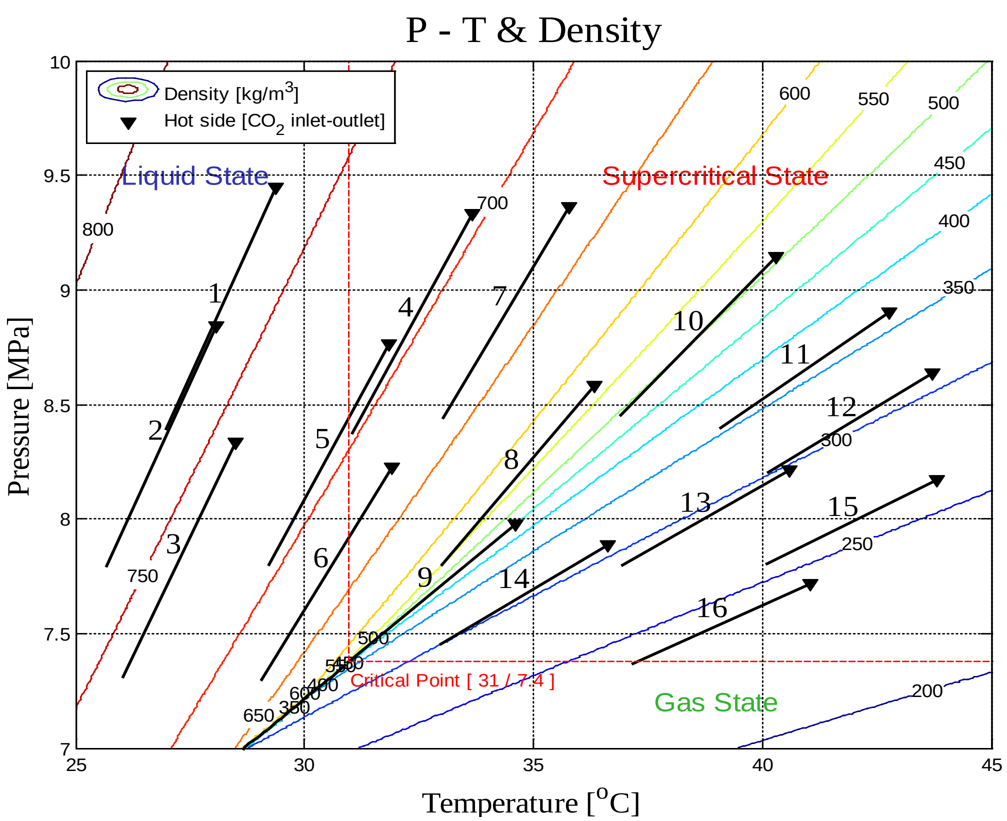

Fig. B. S-CO_2PE pure CO_2 compressor test results (inlet to outlet)

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