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Test rig design for CO2 heat pump system lifting from ultra-low to high temperatures

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Authors		
Author(s) Name	Organisation	E-mail address
Hiroshi Yamaguchi	Doshisha University	hyamaguc@mail.doshisha.ac.jp

Abstract

The working fluid CO2 has been much promoted in recent years due to its environmental friendly and low critical point. Supercritical CO2 has high heat transfer efficiency, which is used to utilize in the system for high efficiency in term of power generation and heat generation. Moreover, the property of CO2 below triple point is utilized in the cooking purposed. The current developments of CO2 applications are reported and discussed in this paper.

Conference Program

The 7th International Workshop of Energy Conversion

(IWEC 2017)

Proceedings

Nantong University

Date: November 25-28, 2017

Venue: Library 508, Nantong University, Nantong, China

Organizers:

School of Mechanical Engineering, Nantong University, China
Science and Technology Association, Nantong University, China
Science and Technology Association, Nantong, China
Nantong Institute of Technology, Nantong, China
Energy Conversion Research Center, Doshisha University, Japan

Sponsors:

Jiangsu Society of Engineering Thermo-physics, China
Nantong Refrigeration Association, China
Nanjing Branch, Daikin (China) Investment Co., Ltd, China
Jiangsu Hao Han Information Technology Co., Ltd, Nantong, China
Nature Science Journal of Nantong University, Nantong University, China

IWEC 2017 Program

Nov.25, Saturday

Register and check in Rosedale Hotel
16:00- Reception desk is at the first floor of Rosedale Hotel.

19:00- **Banquet**

Nov.26, Sunday

Registration

Chairman: Zhang Huali, Nantong University, China
8:30-9:00 Reception desk is in the front of Library 508, Nantong University, and will be open through the whole workshop.

Opening Session

Chairman: Xi Guannan, Nantong University, China
9:00-9:45 Opening Speech I: **Shi Weidong**
President of Nantong University, Nantong, China
Opening Speech II: **Xue Yun**
Representative of Nantong Association for Science and Technology, China
Opening Speech III: **Yamaguchi Hiroshi**
Department of Mechanical Engineering, Doshisha University, Japan

9:45-10:00 **Tea Break**

Session I

Chairman: Niu Xiaodong, Shantou University, China
10:00-10:35 **Invited Talk 1: Wang Ruzhu**
Institute of Refrigeration and Cryogenics, Shanghai Jiaotong University, China
"Renewable Heating and Cooling for Green Buildings"
10:35-11:10 **Invited Talk 2: Yamaguchi Hiroshi**
Department of Mechanical Engineering, Doshisha University, Japan
"Review on the CO_2 Energy Conversion Cycle Applications"
11:10-11:45 **Invited Talk 3: Yuan Yinnan[1], Mei Deqing[2]**
1. School of Energy, Soochow University, China
2. School of Automobile and Traffic Engineering, Jiangsu University, China
"Morphology Features and Carbon Component Analysis of Stratified Diesel Particulates"

12:00-13:00 **Lunch Break**

13:30-14:30 **Visit Fanzeng Art Gallery**

Session II

Chairman: Wang Zhong, Jiangsu University, China
14:30-15:05 **Invited Talk 4: Chen Zhenqian**
School of Energy and Environment, Southeast University, China
"Heat Transfer Enhancement of PCMs with Metal Foams for Solar Energy Storage"
15:05-15:40 **Invited Talk 5: Okubo Masaaki**
Department of Mechanical Engineering, Osaka Prefecture University, Japan
"High Efficient Adsorbed CO_2 Dissociation Using Nonthermal Plasma Flow"
15:40-16:15 **Invited Talk 6: Zhang Peng**
Institute of Refrigeration and Cryogenics, Shanghai Jiaotong University, China
"A Uniform Approach to Describe the Flow and Heat Transfer of Phase Change Material Slurries and the Experimental Verifications"

16:15-16:30 **Tea Break**

Session Ⅲ

Chairman: Cai Yixi, Jiangsu University, China
16:30-17:05 **Invited Talk7: Niu Xiaodong**
College of Engineering, Shantou University, China
"Interface Effect of Multiphase Flow and Nonmagnetic Particles in Magnetic Fluid with Assembly Behavior and Heat Transfer Improvement"
17:05-17:40 **Invited Talk 8: Kuwahara Takuya**
Department of Products Engineering and Environmental Management
Nippon Institute of Technology, Japan
"Fundamental Characteristics of Diesel Particulate Removal from Exhaust Gas using a Magnetic Fluid"
17:40-18:15 **Invited Talk 9: Wang Zhong**
School of Automobile and Traffic Engineering, Jiangsu University, China
"Biomass Energy and Application in the Engine"

18:30- **Banquet**

Nov.27, Monday

Session Ⅳ

Chairman: Sun Ping, Jiangsu University, China
9:00-9:35 **Invited Talk 10: Zhu Ning**
Shizuoka Institute of Science and Technology, Japan
"Development of Hybrid Power Generation System Based on Solar Cell and Thermoelectric Conversion"
9:35-10:10 **Invited Talk 11: Li Mingjun**
School of Mathematics and Computational Science, Xiangtan University, China
"Elliptical Characteristic of Parabolized Stability Equations of Ferrofluid Motion"

10:10-10:25 **Tea Break**

Session V

Presentations:
Chairman: Chen Lin, Tohoku University, Japan
(15 minutes for each presentation, including 5 minutes of answering questions)

1. 10:25-10:40
Visualization Study on Flow Characteristics of Offset Cylinder Arrays in Transitional Flow State
*W.J. Zhang, J. Zhou, J. H. Xu, Y. Rui, G.N.Xi**

2. 10:40-10:55
Preparation of Partially Hydrogenated Biodiesel and Its' Basic Physi-Chemical Properties
*L. Zuo, Q. Zhang, C. Sun, M. Gu, D.Q. Mei**

3. 10:55-11:10
Visualization Experiments on the Flow Characteristics of Circle Cylinders with Unequal Diameters in Tandem Arrangement
*L.R. Tang, L. Zhao, J. Zhou, Y. Rui, G.N. Xi**

4. 11:10-11:25
Characteristics of In-Cylinder Soot Size and Number Density from a Diesel Engine
L. Bai, P.Y. Ni, X.L. Wang, R.N. Li*

5. 11:25-11:40
Effect of the Gap Ratio on Heat Transfer Characteristics in Transitional Flow at Two-Tandem Cylinders Near Wall
*L. Zhou, Y. Shi, M. Yao, G.N. Xi**

6. 11:40-11:55
Study on Transient Heat Transfer for Helium Gas flowing over a Twisted Plate with Different Length
L. Wang, Q.S. Liu, Z. Zhao, K. Fukuda*

11:30-13:00 **Lunch Break**

Session VI

Chairman: Kuwahara Takuya, Nippon Institute of Technology, Japan

13:30-14:05 **Invited Talk 12: Chen Lin**
Department of Aerospace Engineering, Tohoku University, Japan
"On the Backward-facing Step (BFS) as a Basic Separation Flow Model: Recent Trends and Future Directions"

14:05-14:40 **Invited Talk 13: Xie Weian[1], Xi Guannan[2]**
1 School of Mechanical Engineering, Shanghai Jiao Tong University, China
2 Department of Mechanical Engineering, Nantong University, Nantong, China
"Fluid Flow and Heat transfer Characteristics of the Separated and Reattachment Flow over a Step"

14:40-15:15 **Invited Talk 14: Wang Guanbang**
Department of Energy and Resources Engineering,
College of Engineering, Peking University, Beijing, China
"Snowmaking at Ambient Temperature of above 0℃"

15:15-15:30 **Tea Break**

Session Ⅶ

Presentations:
Chairman: Zhu Ning, Shizuoka Institute of Science and Technology, Japan
(15 minutes for each presentation, including 5 minutes of answering questions)

7. 15:30-15:45
Improvement of Heat Transfer at Moderate Rayleigh number of Rayleigh-Bénard Convection with a Particle Inside
M.F. Chen and X.D. Niu*

8. 15:45-16:00
Surface Tension and Surface Energy of Silver Nanowire Ethanol Dispersion
Y. Kishi, H. Yamasaki, B. Jeyadevan, H. Yamaguchi*

9. 16:00-16:15
Influence of Shale Gas Components on Jet Diffusion Flame
R. Jia, Z. Wang, S. Liu., J.H. Yang., Y. Li*

10. 16:15-16:30
Applications of Low-GWP Working Fluids in Mechanical Heat Pumps For Low-Temperature Waste Heat Recovery
*G.B. Wang and X.R. Zhang**

11. 16:30-16:45
Study on the Auto-Ignition Process of Methanol/Biodiesel Blend in Diesel Engine
L. Mei, Z. Wang, R.N. Li, J.H. Yang, L.F. Liu*

12. 16:45-17:00
Flow and Heat Transfer Characteristics in Backward-Facing Steps with the Effect of Expansion Ratios at Low Reynolds Number
*J.H. Shen, J.H Xu, K. Inaoka, G.N. Xi**

13. 17:00-17:15
Bubble Transfer by Magnetic Fluid and Its Application
H. Yamasaki and H. Yamaguchi*

Closing Session

Chairman: Xi Guannan, Nantong University, China
17:15-17:25 Closing Speech: **Yamaguchi Hiroshi**
Department of Mechanical Engineering, Doshisha University, Japan

18:00- **Banquet**

Nov.28, Tuesday

8:00 AM-5:00 PM Tour: Suzhou

Proceeding

Proceedings of IWEC2017
The 7th International Workshop of Energy Conversion
November 25-28, 2017, Nantong, China

Paper No. IWEC2017-0002

REVIEW ON THE CO_2 ENERGY CONVERSION CYCLE APPLICATIONS

H. Yamaguchi*
Energy Conversion Research Center, Department of Mechanical Engineering, Doshisha University, Kyoto, Japan
*corresponding author, E-mail: hyamaguc@mail.doshisha.ac.jp

ABSTRACT

The working fluid CO_2 has been much promoted in recent years due to its environmental friendly and low critical point. Supercritical CO_2 has high heat transfer efficiency, which is used to utilize in the system for high efficiency in term of power generation and heat generation. Moreover, the property of CO_2 below triple point is utilized in the cooling purposed. The current developments of CO_2 applications are reported and discussed in this paper.

INTRODUCTION

To deal with the Global warming and climate change crisis, which have been a great concern in our human history, the prime importance is the greenhouse gas reduction with the highest efforts in technology. The 1997 Kyoto Protocol, the world's first climate change treaty, the Hydrofluorocarbons (HFCs) had been recommended and widely used in previous decades as a working fluid in the industries, instead of other chemical working fluid, due to its no effect to the Ozone Layer Depletion (zero ODP). However, HFCs has very high Impact on Global Warming Potential (high GWP), which directly causes global warming crisis.

In 2015, the 21st session of the Conference of the Parties (COP), so called "Paris Agreement", also suggests reducing the carbon emission at the earliest and handle the increasing of global average temperature to well below 2 °C. To achieve the objective, the natural working fluid such as carbon dioxide (R-744 or CO_2), which is also listed in the required collective control in the Kyoto Protocol has recommended to utilize and use instead of HFCs due to the reasons that CO_2 itself is environmentally friendly, which is 0 and 1 of ODP and GWP, respectively [1].

The critical pressure and critical temperature of CO_2 are 7.38 MPa and 31.1 °C, respectively, which are very much lower than other working fluids as well. So, CO_2 can become as supercritical state easily to use in high efficiency power production cycle. Besides of high pressure and temperature supercritical state, which is used in the power generation cycle, it is mentioned here that a dry ice solid-gas state of CO_2 is also interesting phase for use in a refrigeration system. The refrigeration system can advise a cryogenic temperature below the CO_2 triple point temperature of -56.6 °C and pressure of 0.518 MPa, the detail of CO_2 phase is shown in Fig. 1 as the Moillier (P-h) diagram [2]. Also, CO_2 is classified as non-flammable, non-toxic working fluid and chemical inactive, which are safe to human health and the environment. From these reasons, many attempts have been made to replace ordinary working fluids to CO_2 in energy conversion cycle, in which the Rankine cycle system, solar heat collector and heat pump system, will be introduced and reviewed in this paper.

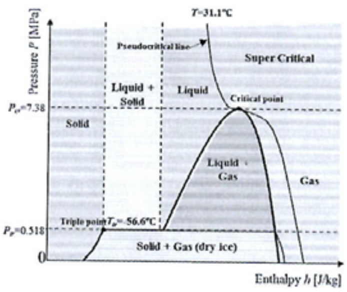

Fig. 1 The Mollier (P-h) diagram of CO_2.

SUPERCRITICAL CO_2 SOLAR RANKINE CYCLE SYSTEM

The ideal Rankine cycle is a practical model of energy conversion from heat to mechanical work. The Rankine cycle gains greater thermal efficiency, in which the cycle provides two phase changes in the cycle process. By taking an account in the environmental pollution, the working fluid CO_2 has received considerable attention to be used in Rankine cycle due to its unique property as the state of supercritical. The supercritical CO_2 solar Rankine cycle system (SRCS), which utilizes solar energy as an input energy and CO_2 as a natural working fluid for the purpose of electric and thermal energy generation has been invented [3].

System Configuration

SRCS consists of main five components and processes, as shown in Fig. 2 and Fig. 3, respectively. The evacuated solar collector, contained CO2 inside, is installed at high pressure side of Rankine cycle, 5 → 1, to absorb the heat from the sun, in which CO2 is heated by solar energy and turns to a supercritical state. The expanded supercritical CO2 drives a turbine and generate electric energy, 1 → 2. After leaving turbine, due to the remaining thermal energy, CO2 is passed through the heat exchanger units, 2 → 3, and heat energy is supplied to appliances on hot water supply. CO2 is further cooled to liquid state in the condenser, 3 → 4. Lastly, mechanical feed pump feeds CO2 back to CO2/methanol heat exchanger with the high-pressure state, 4 → 5, and the cycle recommences. It can be seen that in the cycle process of SRCS both electric and heat energies are generated with high efficiency and environmentally friendly manner, because of unique properties of supercritical CO2.

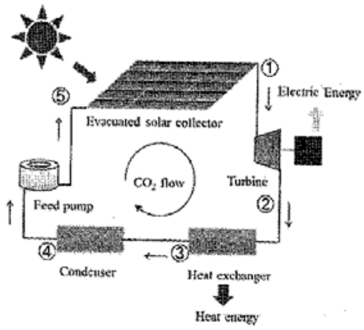

Fig. 2 Schematic diagram on Supercritical Solar CO_2 Rankin cycle system

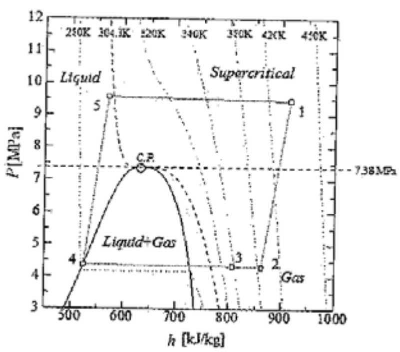

Fig. 3 Pressure-Enthalpy diagram of Supercritical Solar CO_2 Rankin cycle system

The evacuated tube solar collector is well designed to effectively absorb solar radiations (shown in Fig 4). The inside tube coated with a selective solar absorber coating, which can collect short wave radiation from sun and prevent the loss of long wave radiation from tube [4].

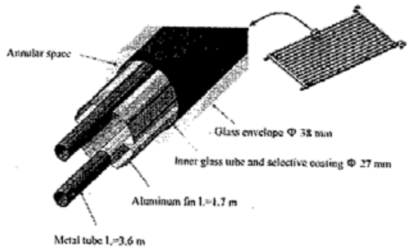

Fig. 4 A sketch of the evacuated solar collector used in Supercritical Solar CO_2 Rankin cycle system

In the SRCS, the feed pump work, the mass flow rate of CO_2 has much significant to the total power output, and the total cycle efficiency of the cycle due to the high rate of energy absorb in CO_2. In this study, to enhance power output, the relationship between mass flow rate and power generation was investigated.

System Evaluations

The amount of energy output at turbine (Q_t) and heat exchanger (Q_h) are estimated by equation (1) and (2), respectively,

$$Q_t = \dot{m}(h_1 - h_2) \tag{1}$$

$$Q_h = \dot{m}(h_2 - h_3) \tag{2}$$

The total energy output (Q) can be obtained from equation (3),

$$Q = Q_t + Q_h \tag{3}$$

by $\dot{m}$ is mass flow rate. The results of total energy output and temperature at outlet of solar collector of the SRCS with two different mass flow rates with approximately 0.63 kW/m² of solar radiation are shown at Fig. 5. It can be obviously seen that total energy output increases with increasing of temperature at outlet of solar collector. In normal thermodynamic cycle, while increase the mass flow rate of the working, the temperature gets decrease. However, in SRCS, due to the unique properties of CO_2, the temperature increases along with increasing of mass flow rate of CO_2. The reasons may be from the low specific heat of CO_2 in supercritical state in high mass flow rate condition, which occurs due to the expanding of supercritical CO_2 in the limited space of the system (i.e. in evacuated solar collector). In the same condition of solar radiation, the high mass flow rate of CO_2 can induce high system efficiency compared with low flow rate, in which can be confirmed by the equations shown below.

The system efficiency (η) of the SRCS can be found from equation (4),

$$\eta = \frac{(h_1 - h_3) - (h_5 - h_4)}{(h_1 - h_5)} \tag{4}$$

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Results and discussions

From experimental results of energy output shown in Fig. 5 and calculated system efficiency from equation (4), it can be concluded that in the case of high flow rate condition, 0.63 kg/min, the values can be found much higher than low flow rate, condition, 0.35 kg/min, which are 17.3% and 1.34 kW, and 13.3% and 0.547 kW of energy output and system efficiency, respectively. It is obviously confirmed that higher mass flow rate gives higher energy output. In which, the reasons may due to the higher energy collection in evacuate solar collector part, h1-h5. Thus, the SRCS can produce higher energy by increasing mass flow rate. From the estimation of high mass flow rate, the amount of total energy can be produced is 1.85 kW.

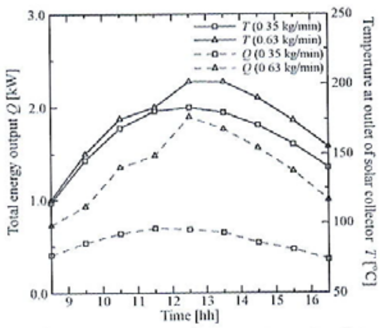

Fig. 5 Total energy output and temperature at outlet of solar collector with difference mass flow rate (0.35 kg/min and 0.63 kg/min)

The efficiency of SRCS is obtained from equation (4), which is about 35 %, while to compare the system efficiency with the ordinary photovoltaic system, which is around 20 % [5]. It can be concluded here that SRCS shows much efficiency and sustainability in energy production.

SOLAR WATER HEATER

Since the amount of energy that used to supply hot water in our society is large, the solar water heater has been used and studied extensively in the past time and has been put to practical use for many years. However, most of the solar water heaters in the market use water as a working fluid, which has low specific heat capacity and result in low efficiency. Natural circulation model of heat recovery system using supercritical CO_2 was developed, as well as the thermal properties and flow behaviors were experimentally investigated. It is highly expected that the system provides a possibility to circulate CO_2 without any mechanical pumps with the high heat recovery efficiency achieved. In this system aiming to supply hot water by exchanging the heat from high temperature CO_2 to low temperature water in this system.

Experimental set-up and principle

Fig. 6 shows the schematic diagram of solar heat recovery system. Solar heat recovery system is included with CO_2 closed loop structure consisting of solar collector and heat exchanger, and water opened loop structure consisting of heat exchanger and water flow cycle. Liquid CO_2 absorbs heat from the sun, in which CO_2 is heated by solar energy and turns to a supercritical state. After leaving the solar collector, CO_2 is passed through the heat exchanger units and turns to a liquid CO_2. The density difference induces by large density difference in the system. The system can recover the solar heat without mechanical feed pump. In winter season in Kyoto, Japan, the ambient temperature is around 10 ℃ which system cannot operate due to taking many time to absorb heat by CO_2. In this study, a heater is installed before the solar collector to pre-heat CO_2.

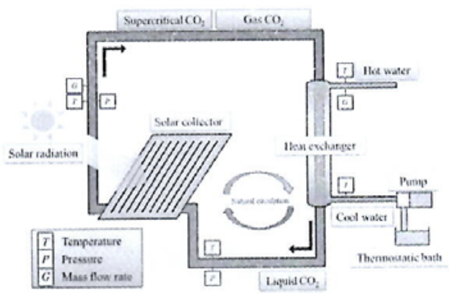

Fig. 6 Schematic of solar heat recovery system[6]

Results and discussions

In the present study, the solar collector is set declined with an angle of 30° with respect to the horizontal level. The following items were mainly measured: the solar radiation, the CO_2 fluid pressures at the inlet and outlet of the solar collector, the CO_2 fluid temperatures at the inlet and outlet of the heat exchanger, the mass flow rates of CO_2 fluid and water, and the water temperature at the inlet and outlet of the solar collector.

The experimental result of amount of solar heat and mass flow rate of CO_2 was carried out from 8:00 to 16:00, shown in Fig. 7. In this experiment, CO_2 was heated by heater from 8:00 to 8:15. This is caused the mass flow rate of CO_2 rapidly increased from 8:00 to 8:15 before sufficient solar radiation is obtained. When solar radiation increases from 8:00 to 12:00, it is found that the mass flow rate of CO_2 achieves more than 10 kg/h. The mass flow rate of CO_2 decreased after decreasing of solar radiation (12:00 to 16:00) by CO_2 absorbing remained heat in the solar collector.

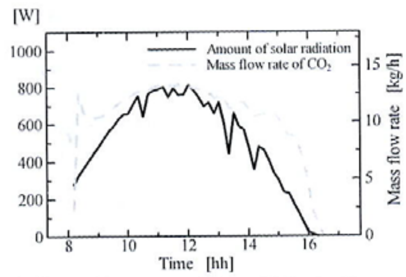

Fig. 7 Variations of the measured data with the test time, which include solar radiation and CO_2 flow rate

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Collected heat quantity in the solar collector and recovered heat quantity by the water flow in the heat exchanger are shown in Fig. 8. It is observed from Fig. 8, most of the time during the test period, the collected heat quantity is relatively stable at a value of 497 W and the value of the recovered heat quantity is calculated at 406 W. These values of the collected and recovered heat quantity are the same as the average values in summer season in Japan.

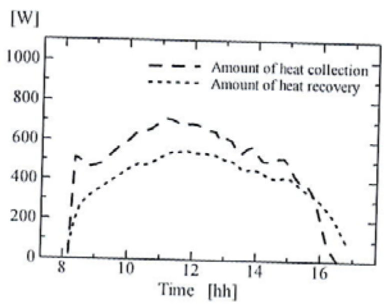

Fig. 8 Variations of the performance parameters with the test time, which include collected heat quantity and recovered heat quantity

HEAT PUMP SYSTEM

The CO_2 ultra-low temperature cascade refrigeration system with dry ice sublimation is composed of two CO_2 refrigeration compression cycles, namely high pressure cycle (HPC) and low pressure cycle (LPC), each of which is composed of condensers, an expansion valve, an evaporator and a compressor. The two CO_2 refrigeration cycles are arranged in cascade trough a brine channel in order to form dry ice (solid-gas two-phase flow) in the evaporator of the LPC. A feasibility study of the system operation bellows the CO_2 triple-point temperature of -56.6 °C has been performed [7], and the system has shown the capability of achieving continuous and stable operation at a cryogenic temperature of approximately -62 °C at the suitable experimental condition. However, the dry ice blockage is sometimes generated. The dry ice blockage is phenomena that accumulated dry ice particles (dry ice sedimentation) block the cross section of the evaporator and causes the vacuum operation behind the blockage in the evaporator. In this report, a tapered evaporator/sublimator is newly designed and installed into the existed CO_2 ultra-low

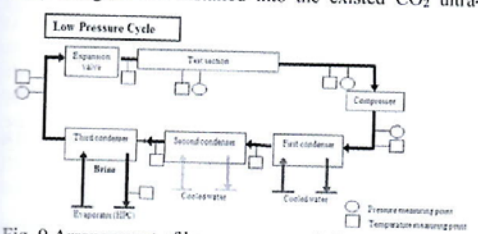

Fig. 9 Arrangement of low pressure cycle in experimental CO_2 ultra-low temperature cascade refrigeration system [8]

temperature cascade refrigeration system in order to investigate the heat transfer characteristics and its associated system performances. [8].

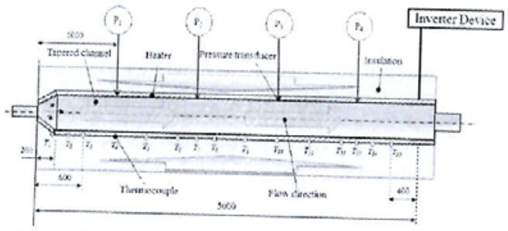

Fig. 10 Assembly of tapered evaporator (test section) [8]

Fig. 9 and Fig. 10 depict the detailed schematic diagrams of the LPC in the CO_2 ultra-low temperature cascade refrigeration system and the tapered evaporator (test section) in the LPC, respectively. As shown in Fig. 9, the LPC is mainly composed of three condensers, an expansion valve, an evaporator (test section) and a compressor, and the detail schematic diagram of the newly designed evaporator (the test section) is depicted in Fig. 10. The evaporator is mainly composed of a tapered channel, heater and thermal insulation.

In this study, the system performance of the LPC is focused that the following parameters are adjusted and considered in the present study; the opening of the expansion valve of 15 mm, the heating quantity of the heater of 1200 W (1910 W/m² heat flux), the rotation speed of the compressor of 60 Hz and the condensation temperature of -20 °C, -25 °C, -30 °C. It should be noted that the condensation temperature is the CO_2 temperature at the inlet of the expansion valve, which is adjusted by controlling the temperature of the third condenser connected with the HPC. All measured data are transferred into computer through a signal distributor and data logger.

In the present study, the local heat transfer coefficient along the evaporator is investigated at various condensation temperatures as shown in Fig. 11, the lateral axis shows the position from the inlet of the evaporator and the longitudinal axis represents the local heat transfer coefficient. The local heat transfer coefficient can be calculated by the equation (5):

$$h_x = \frac{q}{T_w - T_{in}} \qquad (5)$$

where q is the heat flux, T_w is the inner wall temperature of the tube and Tin is the CO_2 temperature in the evaporator. The CO_2 temperature in the evaporator T_{in} is estimated by the energy balance between the inlet and outlet of the evaporator. As shown in Fig. 11, the local heat transfer coefficient decreases when the condensation temperature decreases that the local heat transfer coefficient decreases in the range of 0-3000 mm, and then increases in the range of 3000-4000 mm, while it decreases in the range of 4000-5000mm.

Results and Discussions

In the present study, as described in the previous section, the newly designed tapered evaporator/sublimator is actually installed in the existed CO_2 ultra-low temperature cascade

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refrigeration system, and the heat transfer characteristics are examined together with testing the system performance of the CO_2 ultra-low temperature cascade refrigeration system.

Fig. 11 shows the local heat transfer coefficient in the tapered evaporator/sublimator. As shown in Fig.7, the local heat transfer coefficient decreases when the condensation temperature decreases, which means that the refrigeration work of the evaporator/sublimator decreases with the decreases of the condensation temperature. With respect to the reason for the variation of the heat transfer coefficient, it can be considered that the mass flow rate decreases with the decrease of the

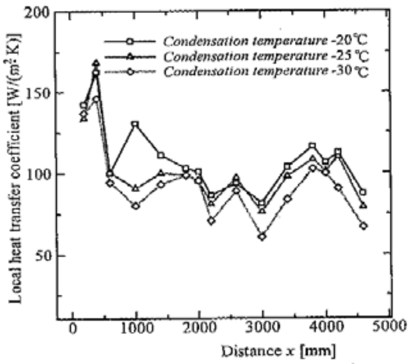

Fig. 11 Calculated local heat transfer coefficient of CO_2 flow at various condensation temperatures along evaporator in the LPC) [8]

condensation temperature. According to the CO_2 Mollier diagram as shown in Fig.1, when the condensation temperature decreases along with the wet saturated steam curve, the thermodynamic cycle of the LPC is shifted toward lower pressure and enthalpy side (this thermodynamic cycle variation depending on the condensation temperature can be found in [9].

Resultantly, it can be concluded that the effective evaporator temperature decreases with decrease of the condensation temperature. This is due to the condensation pressure which decreases along the saturated steam curve. The lowest cryogenic refrigeration temperature of -66.4 °C was achieved at the condensation temperature of -30 °C without the system operation failed. From these results, it was found that the heat transfer in the evaporator was increased by covering dry ice in inner wall of pipe in the evaporator.

CONCLUSIONS

(1) The thermodynamic system analyses based on the measured data show that the CO_2-based cycle can achieve stable outputs of power efficiency of 17.3 % and energy output of 1.34 kW. The study shows the potential of the application of the CO_2-based cycle powered by solar energy.

(2) By installing the pre-heater in the solar water heater system, even in winter season, this natural convection flow is used to achieve solar to thermal convection process. The measured data show that a recovered heat quantity of 406 W can be obtained in the heat exchanger.

(3) With replacing the evaporator/sublimator from the previous sudden expansion to the new tapered type, it is found that the system has the capability of achieving the ultra-low temperature of -66.4 °C continuously the achieved refrigeration temperature with the tapered evaporator/sublimator is 3.5 °C lower than that of the sudden expansion evaporator/sublimator which is installed in the previous system.

ACKNOWLEDGEMENT

A part of this report has been funded by HighEFF - Centre for an Energy Efficient and Competitive Industry for the Future. The authors gratefully acknowledge the financial support from the Research Council of Norway and user partners of HighEFF, an 8 year Research Centre under the FME-scheme (Centre for Environment-friendly Energy Research, 257632/E20).

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7

Presentation

Review on the CO_2 Energy Conversion Cycle Applications

International Workshop of Energy Conversion
IWEC2017
November 25-28, 2017
Nantong University, China

Hiroshi Yamaguchi, Ph.D.

Energy Conversion Research Center

Professor

Department of Mechanical Engineering

Fluid Mechanics Laboratory, Doshisha University, Japan

Research background of CO_2 as a working fluid

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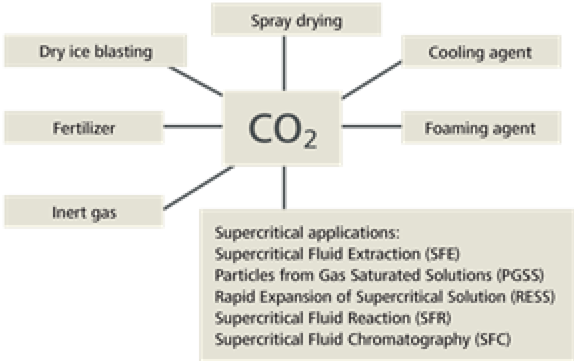
CO_2 for various industrial applications 3

Industrial applications of CO_2 and $scCO_2$



CO_2 is one of the so-called "natural" refrigerants, a group that includes ammonia, hydrocarbons such as propane and butane, and water. Although CO_2 is necessary for all life on earth, it is also **a greenhouse gas that can bring about environmental changes if its atmospheric concentration changes.**

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Why CO_2? 4

Advantage of CO_2 as working fluid

Fluid name	ASHRAE No.	Formula	Critical temp.(°C)	Critical press.(bar)	ASHRAE Level for safety[*4]	ODP rel.R11	GWP rel.CO_2
Carbon dioxide	R744	CO_2	30.98	73.8	A1	0	1
Water	R718	H_2O	373.89	22.1	A1	0	-
Hydrofluorocarbon	R134a	CH_2FCF_3	101.1	40.7	A1	0	1300
Butane	R600	C_4H_{10}	152	37.9	A3	0	0
Isobutane	R600a	C_4H_{10}	134.7	36.4	A3	0	<20
Sulfur Dioxide	R764	SO_2	157.5	78.8	B1	0	-
Ammonia	R717	NH_3	132.89	112.8	B2	0	<1
Sulfur Hexafluoride	R-7146	SF_6	45.56	37.6	?	?	239000
Perfluoropropane	R218	C_3F_8	71.89	26.8	?	0	7000
Zeotropic mixture	R407C	R32/R125/R134a(23/25/52)	87.3	48.2	A1	0	1600
Azeotropic mixture	R500	R12/R152a (73.8%/26.2%)	102.1	41.7	A1	0.605	7870

[*4] In ANSI/ASHRAE Standard 15-1992, refrigerants are classified according to the hazard involved in their use. Group A1 refrigerant are the least hazardous, Group B3 the most hazardous.

A: low toxicity , B: high toxicity , 1: uninflammable , 2: mild flammable , 3: extremely flammable

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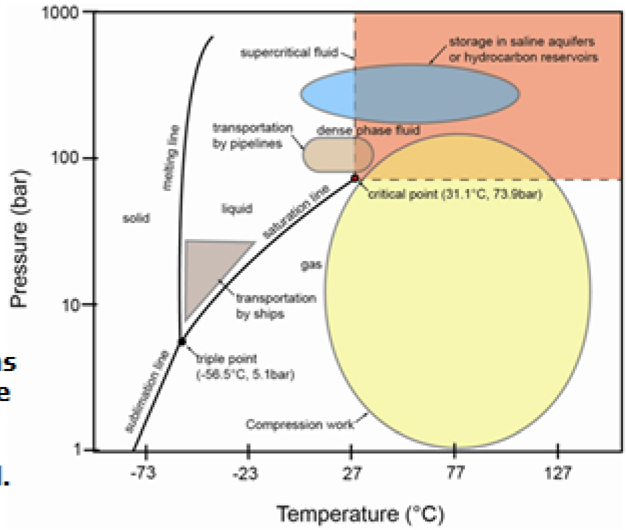
A pressure-temperature diagram

5

- the sublimation curve
- the fusion curve
- the vaporization curve
- the triple point
- the critical point



The pressure and temperature domains superimposed on the CO_2 phase diagram with various operations purposed.

Paul, S., Shepherd, R., & Woollin, P. (2012, June). Selection of materials for high pressure CO2 transport. In *The Third International Forum on the Transportation of CO2 by Pipeline, Newcastle, UK.*

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Virial Coefficient of Carbon Dioxide

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Most General Equation of State

7

- An equation of state exists relating pressure, molar or specific volume, and temperature for any pure homogeneous fluid in equilibrium states.
- An equation of state may be solved for any one of the three quantities P, V, or T as a function of the other two.
- Example: $\boxed{dV = \left(\frac{\partial V}{\partial T}\right)_P dT + \left(\frac{\partial V}{\partial P}\right)_T dP}$

Volume expansively: $\beta \equiv \frac{1}{V}\left(\frac{\partial V}{\partial T}\right)_P$ Isothermal compressibility: $\kappa \equiv -\frac{1}{V}\left(\frac{\partial V}{\partial P}\right)_T$

$$\boxed{\frac{dV}{V} = \beta dT - \kappa dP}$$

- For incompressible fluid, both β and κ are zero.
- For liquids β is almost positive (liquid water between 0°C and 4°C is an exception), and κ is necessarily positive.
- At conditions not close to the critical point, β and κ can be assumed constant $\boxed{\ln\frac{V_2}{V_1} = \beta(T_2 - T_1) - \kappa(P_2 - P_1)}$

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Virial equation of state

8

- PV along an isotherm:
- $PV = a + bP + cP^2 = a(1 + B'P + C'P^2 + D'P^3 + ...)$
  - The limiting value of PV as $P \rightarrow 0$ for all the gases:
  - $(PV)^* = a = f(T)$
  - $(PV)^* = a = RT$, with R as the proportionally constant.
  - Assign the value of 273.16 K to the temperature of the triple point of water: $(PV)_t^* = R \times 273.16$
- Ideal gas:
  - the pressure ~ 0; the molecules are separated by infinite distance; the intermolecular forces approaches zero.
  - $R = \frac{(PV)_t^*}{273.16} = 83.1447\ \frac{cm^3\ bar}{mol\ K}$

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Virial equation of state (cont.)

- **CO_2 compressible factor calculated from the Angus EOS** (Angus et al. 1976)

The compressibility factor for CO_2 was calculated from the state equation;

$$P = \rho RTZ$$

in which Z (Angus EOS) is given as

$$Z = 1 + \frac{\rho}{\rho_c} \sum_{i=1}^{6} \sum_{j=1}^{9} a_{ij} \left(\frac{\rho}{\rho_c} - 1\right)^i \left(\frac{T_c}{T} - 1\right)^j$$

$\rho_c = 468$ kg/m³ and $T_c = 304.2$K, which are CO_2 critical density and critical pressure, respectively.

Coefficients for a_{ij}

i \ j	0	1	2	3	4	5	6
0	-0.725854437	-1.68332974	0.259587221	0.376945574	-0.67075527	-0.871456126	-0.149156928
1	0.447869183	1.26050691	5.96957049	15.4645885	19.4449475	8.64880497	0
2	-0.172011999	-1.83458178	-4.61487677	-3.81121926	3.6171249	4.92265552	0
3	0.004463049	-1.76300541	-11.1436705	-27.8215446	-27.168572	-6.42177872	0
4	0.255491571	2.37414246	7.50925141	6.61133318	-2.4286321	-2.57944032	0
5	0.05946873	1.16974683	7.4370641	15.0646721	9.57496845	0	0
6	-0.147960001	-1.69233071	-4.68219937	-3.12517448	0	0	0
7	0.013671044	-0.10049233	-1.63653806	-1.87082988	0	0	0
8	0.039228458	0.441503812	0.88674197	0	0	0	0
9	-0.01198721	-0.084605195	0.046456437	0	0	0	0

Thermodynamic Tables Project. (1976). *International Thermodynamic Tables of the Fluid State: Carbon Dioxide, 1973.* S. Angus, B. Armstrong, & K. M. de Reuck (Eds.). Pergamon press.

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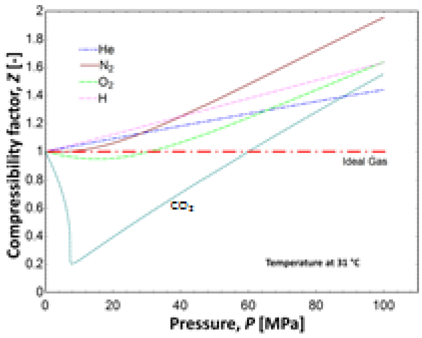
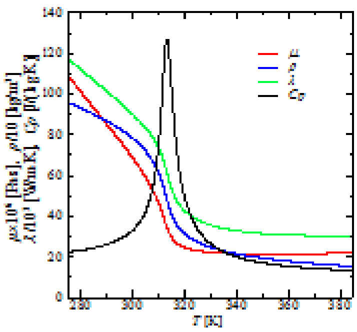
CO_2 Properties

In comparison, the Angus EOS has more advantages compared with Peng-Robinson EOS in accuracy to represent measured CO_2 data.



The compressibility factor (Z) of various working fluids with pressure.



Thermophysical property of CO_2 at 9.0MPa

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CO_2 Thermophysical Property

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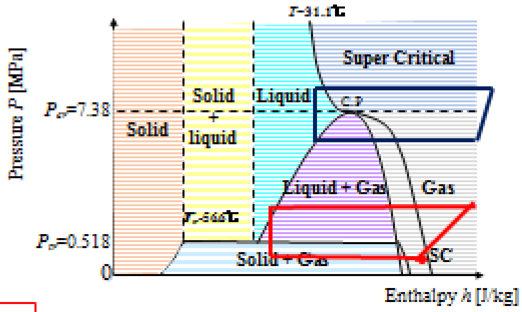
CO_2 (R744)

Molecular weight	: 44.01
Thermal conductivity	: λ=1.607 × 10-2 [W/(m·K)]
Viscosity	: μ = 1.473 × 10-5 [Pa·s]
Heat capacity	: Cp = 845.3 [J/(kg·K)]
Density	: ρ = 1.8389 [kg/m3]
Specific ratio	: γ = 1.297 [-]
Prandtl number	: Pr = 0.7748 [-]
Sublimation temperature	: T_{sub} = 194.7 [K]

(at standard conditions 101.3kPa, 293.15K)



CO_2 Critical point
P_c = 7.38MPa
T_c = 31.4℃

CO_2 Triple point
P_{tr} = 0.518MPa
T_{tr} = −56.6℃

P-h diagram

CO_2 is a good refrigerant

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CO_2 Research for Energy Conversion

12

CO_2 cascade heat pump system with dry-ice sublimation

Solar water heater using supercritical CO_2

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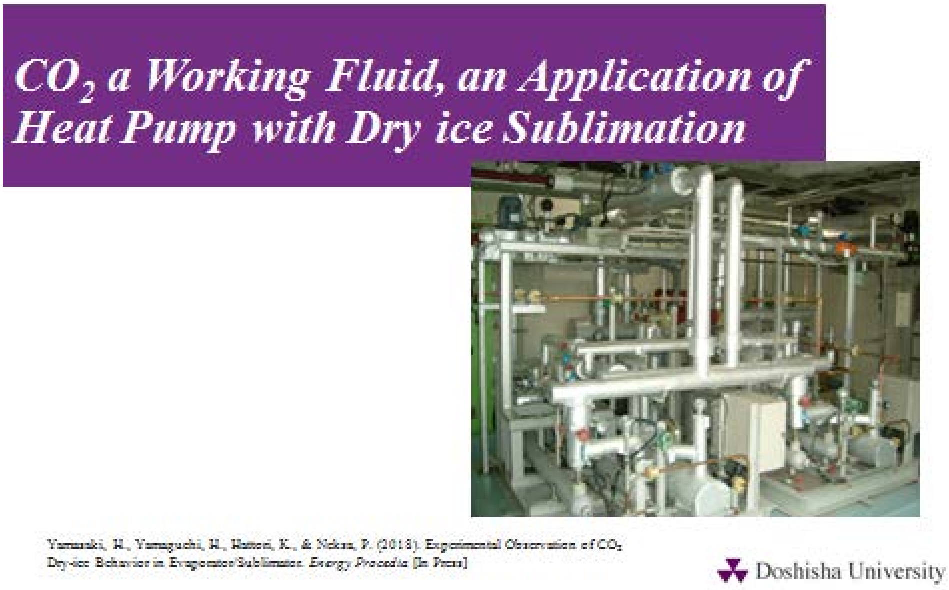
CO_2 a Working Fluid, an Application of Heat Pump with Dry ice Sublimation
Yamasaki, H., Yamaguchi, H., Hattori, K., & Neksa, P. (2018). Experimental Observation of CO_2 Dry-ice Behavior in Evaporator/Sublimator. Energy Procedia [In Press]
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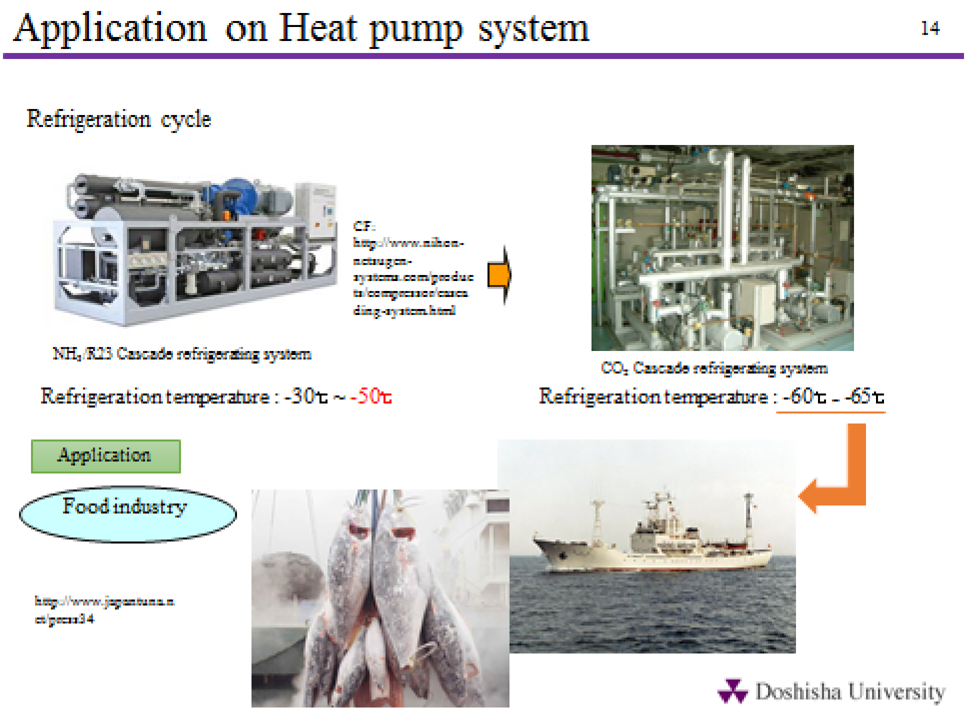
Application on Heat pump system
14
Refrigeration cycle
CF: http://www.nihon-netsugen-systems.com/products/compressor/cascading-system.html
NH_3/R23 Cascade refrigerating system
Refrigeration temperature : -30℃ ~ -50℃
CO_2 Cascade refrigerating system
Refrigeration temperature : -60℃ - -65℃
Application
Food industry
http://www.japantuna.net/press34
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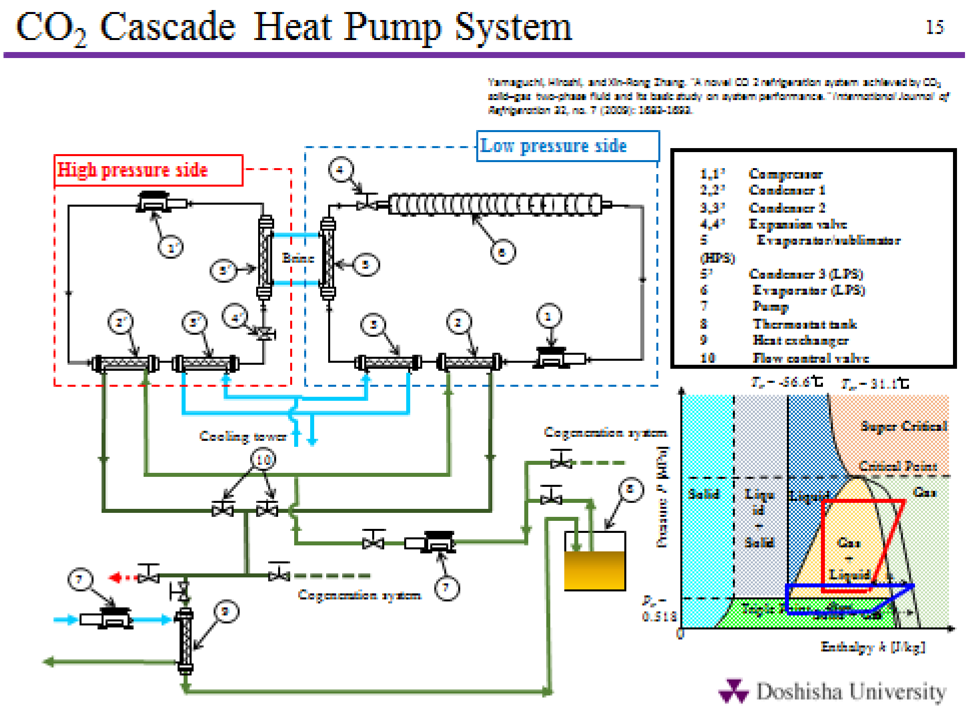
CO_2 Cascade Heat Pump System
15
Yamaguchi, Hiroshi, and Xin-Rong Zhang. "A novel CO 2 refrigeration system achieved by CO_2 solid–gas two-phase fluid and its basic study on system performance." International Journal of Refrigeration 32, no. 7 (2009): 1683-1693.
High pressure side
Low pressure side
Brine
Cooling tower
Cogeneration system
Cogeneration system
1,1' Compressor
2,2' Condenser 1
3,3' Condenser 2
4,4' Expansion valve
5 Evaporator/sublimator (HPS)
5' Condenser 3 (LPS)
6 Evaporator (LPS)
7 Pump
8 Thermostat tank
9 Heat exchanger
10 Flow control valve
T_c = -56.6℃
T_{cr} = 31.1℃
Super Critical
Critical Point
Solid
Liquid + Solid
Liquid
Gas
Gas + Liquid
Triple Point
Pressure P [MPa]
Enthalpy h [J/kg]
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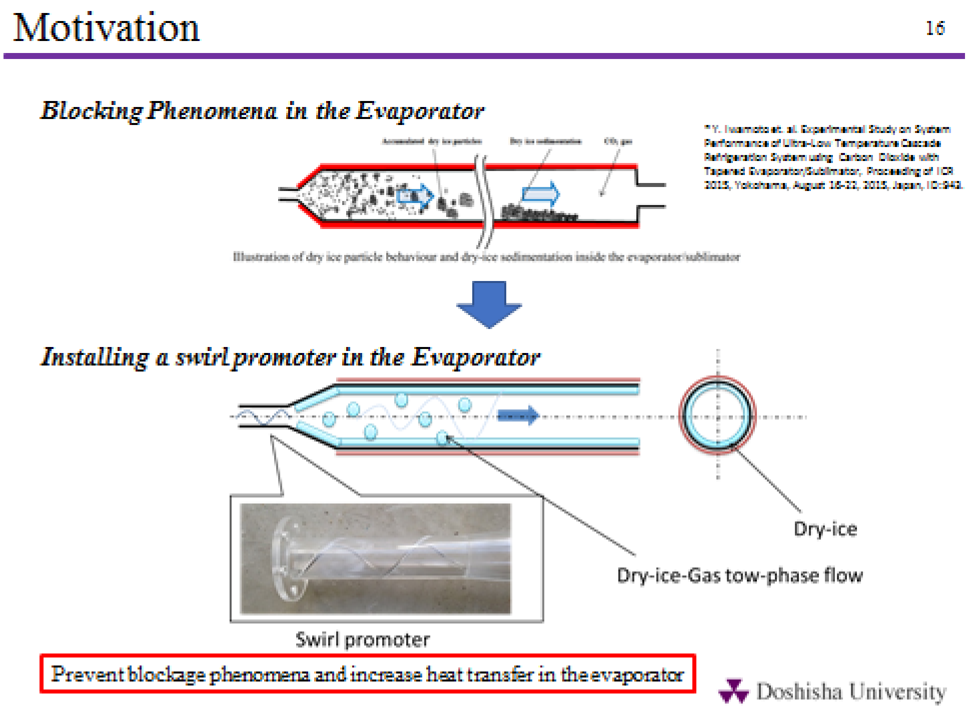
Motivation
16
Blocking Phenomena in the Evaporator
*Y. Iwamoto et. al. Experimental Study on System Performance of Ultra-Low Temperature Cascade Refrigeration System using Carbon Dioxide with Tapered Evaporator/Sublimator, Proceeding of ICR 2015, Yokohama, August 16-22, 2015, Japan, ID:943.
Illustration of dry ice particle behaviour and dry-ice sedimentation inside the evaporator/sublimator
Installing a swirl promoter in the Evaporator
Dry-ice
Dry-ice-Gas tow-phase flow
Swirl promoter
Prevent blockage phenomena and increase heat transfer in the evaporator
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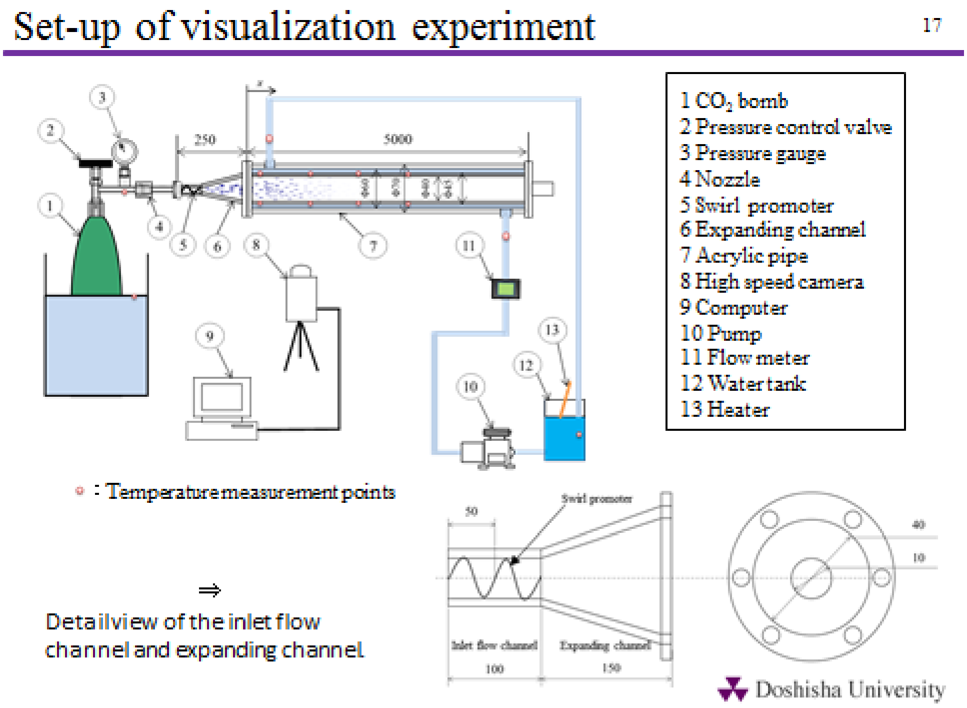
Set-up of visualization experiment
17
1 CO2 bomb
2 Pressure control valve
3 Pressure gauge
4 Nozzle
5 Swirl promoter
6 Expanding channel
7 Acrylic pipe
8 High speed camera
9 Computer
10 Pump
11 Flow meter
12 Water tank
13 Heater
250
5000
: Temperature measurement points
⇒
Detail view of the inlet flow channel and expanding channel
Swirl promoter
50
Inlet flow channel
Expanding channel
100
150
40
10
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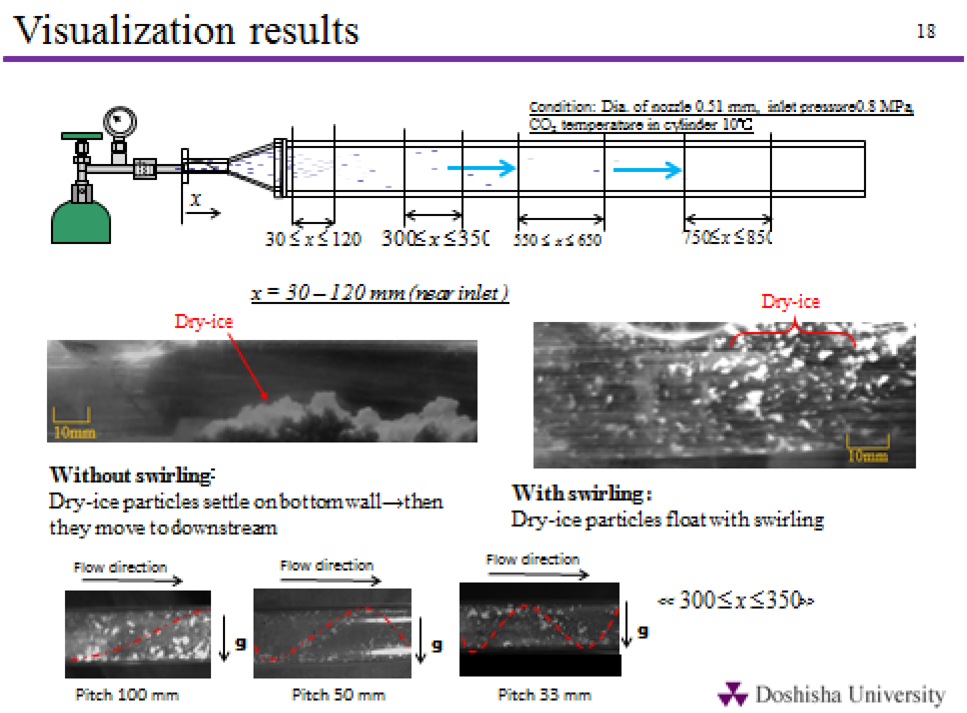
Visualization results
18
Condition: Dia. of nozzle 0.51 mm, inlet pressure0.8 MPa, CO2 temperature in cylinder 10℃
x
30 ≤ x ≤ 120
300≤ x ≤350
550 ≤ x ≤ 650
750≤x ≤850
x = 30 – 120 mm (near inlet)
Dry-ice
Dry-ice
10mm
10mm
Without swirling:
Dry-ice particles settle on bottom wall→then they move to downstream
With swirling:
Dry-ice particles float with swirling
Flow direction
Flow direction
Flow direction
g
g
g
<< 300≤x≤350 >>
Pitch 100 mm
Pitch 50 mm
Pitch 33 mm
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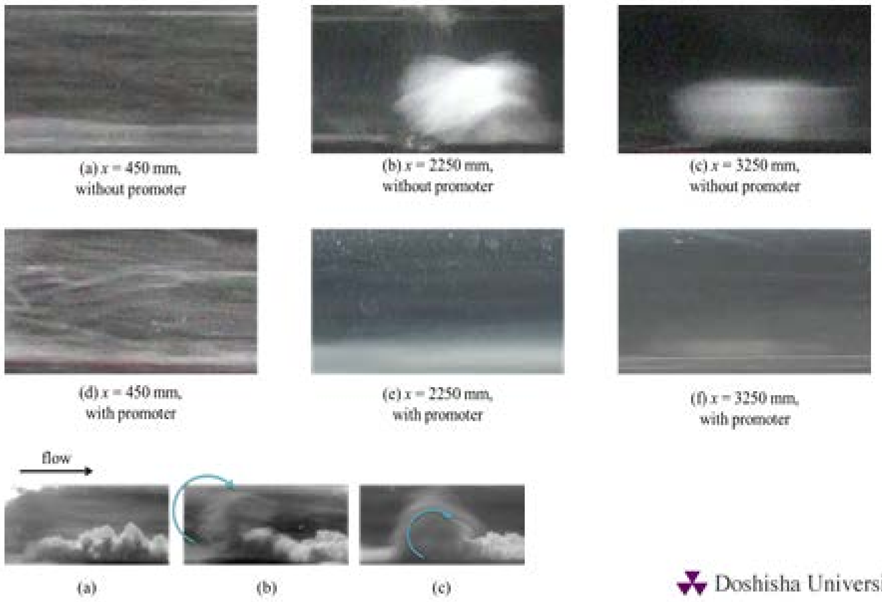
Visualization results at $P = 1.0$ MPa

19



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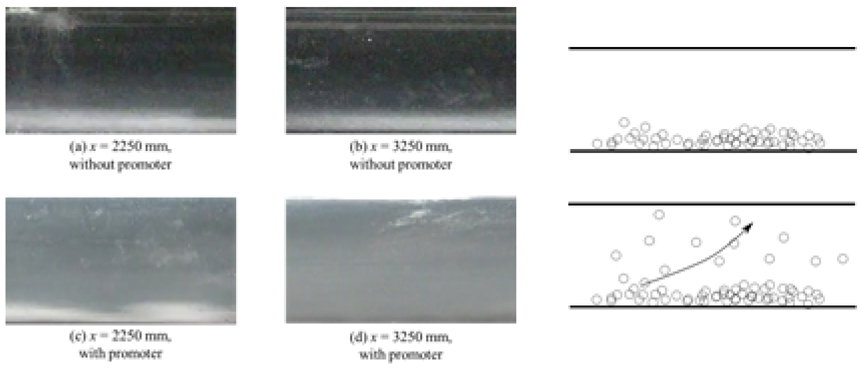
Visualization results at $P = 1.5$ MPa

20



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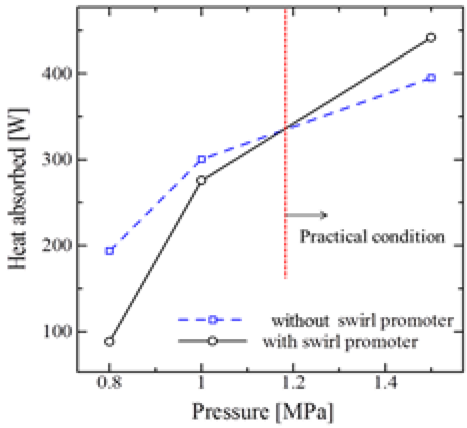
Measurement result of the heat absorbed 21



Heat absorbed at various pressure conditions.

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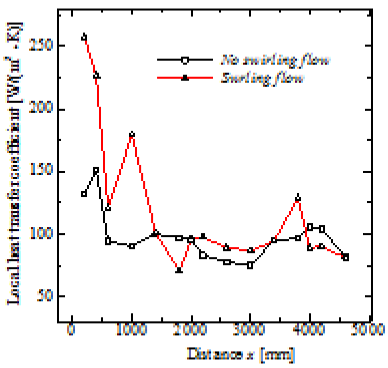
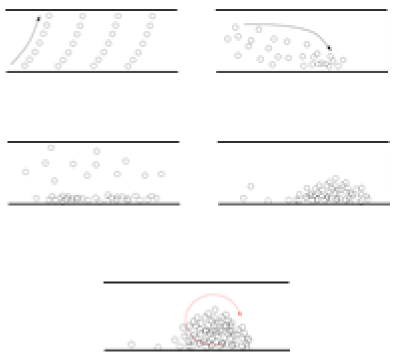
Heat transfer coefficient in the heat pump system 22

Experimental Cond.: Heat flux 1909.9 w/m², Compressor 60 Hz, Condensation temperature -20˚C, pressure ratio 6.21





Calculated local heat transfer coefficient of CO_2 flow with and without swirl promoter along evaporator/sublimator in the heat pump system

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Solar Water Heater using Supercritical CO_2 as a Working Fluid

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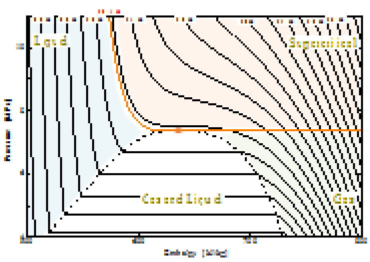
Background

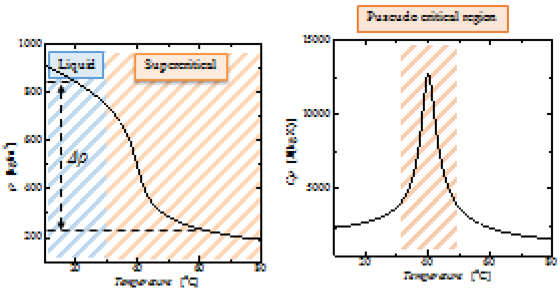
24

Carbon dioxide (CO_2)

Critical pressure 7.38 MPa Critical temperature 31.0 ℃



Mollier diagram (Pressure – Enthalpy)



Variation of CO_2 properties at 9 MPa

- By using CO_2 as a working fluid in the solar heat recovery system, CO_2 natural circulation at low temperature (at 31.0℃) can be achieved due to the CO_2 phase change
- The efficiency of the system can be improved by increasing the heat transfer efficiency in the phase change process.

High efficiency heat recovery and heat transfer are possible to operate without mechanical feed pump

PROPATH v13.1

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Background (cont.) 25

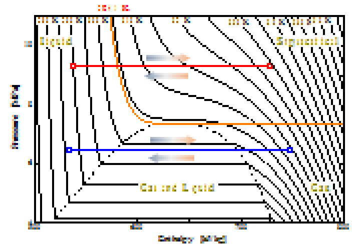
Critical pressure and temperature of CO_2 and H_2O [1]

	p_{cr} [MPa]	T_{cr} [K]
CO_2	7.38	304.1
H_2O	22.1	647.1

CO_2 can be realized in a ambient temperature with a relatively low pressure condition

CO_2 maximum temperature in difference phase state condition

Phase state	Pressure [MPa]	Maximum CO_2 temperature [℃]
Liquid to Supercritical	9.33	55.4
Liquid to Gas	6.49	40.8



- Phase state change from liquid to gas also can be induced natural convection and collect heat
- CO_2 maximum temperature is decreases due to using heat as boiling latent heat when phase state change from liquid to gas

Supercritical state is suitable for water heater system

PROPATH v13.1

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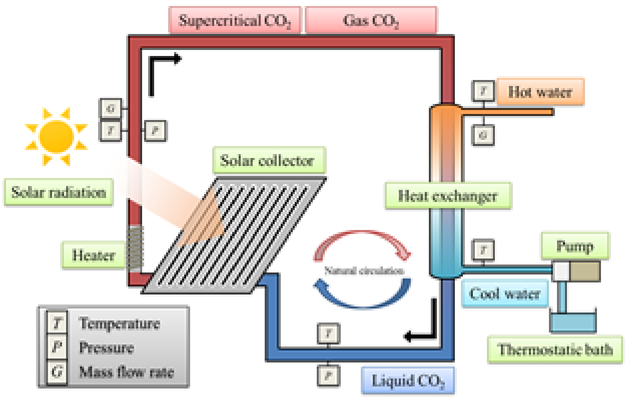
Experimental setup and principle 26



Experimental setup

Solar heat recovery system

To develop high efficiency solar heat recovery system and investigate thermo-fluid characteristic of supercritical CO_2

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Experimental method and conditions

27

Experimental conditions of the system

Mass flow rate of water [kg/h]	20.0
Filling amount of CO_2 [kg]	3.6
Collector tilt angle [°]	30

Experimental place

Doshisha Univ., Kyotanabe city, Kyoto, Japan (E135.5, N34.5)

Experimental conditions of the weather

Date [year/month/day]	Average ambient temperature [℃]	Maximum ambient temperature [℃]	Average solar radiation [W]	Cooling water temperature [℃]
2017/10/26	14.3	22.4	679 (926)	20.9

Evaluation method

Amount of heat collection [W]	$Q_c = G_c \times (h_{out} - h_{in})$
Amount of heat recovery [W]	$Q_w = G_w \times c_p (T_{out} - T_{in})$
Heat collection efficiency [-]	$\eta_c = Q_c / Q_s$
Heat recovery efficiency [-]	$\eta_w = Q_w / Q_s$

G_c [kg/h]: Mass flow rate of CO_2
G_w [kg/h]: Mass flow rate of water
h_{in} [J/kg]: CO_2 enthalpy of collector inlet
h_{out} [J/kg]: CO_2 enthalpy of collector outlet
T_{in} [K]: Water temperature of heat exchanger inlet
T_{out} [K]: Water temperature of heat exchanger outlet
c_p [J/(K·kg)]: Specific heat of CO_2
Q_s [W]: Amount of solar radiation

Properties

The properties is calculated using PROPATH v13.1

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Experimental results

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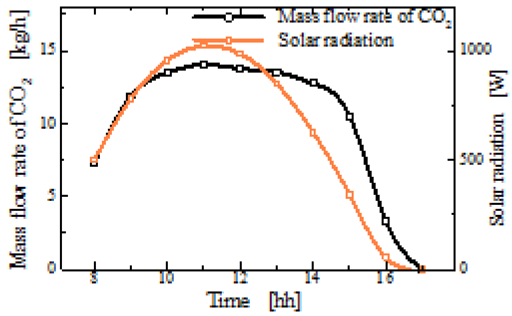
Autumn (2017/10/26)



Mass flow rate of CO_2 start driving after experiment is started

Maximum mass flow rate of CO_2 is 14.2 kg/h when culmination time

Mass flow rate of CO_2 increases with increasing solar radiation and stops when sun starts setting

CO_2 natural circulation is induced by solar radiation

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Experimental results (cont.)

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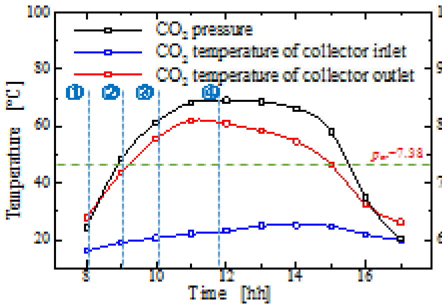

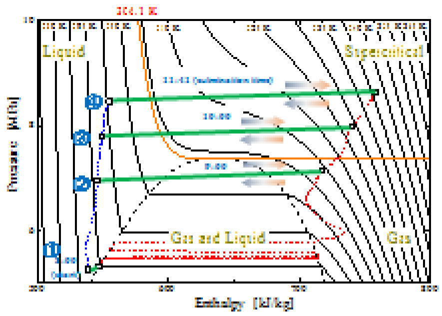

① CO_2 state is liquid state (temperature and pressure are 17.5 °C and 5.3 MPa)
② CO_2 phase state change state is liquid state to gas state
③ ④ CO_2 pressure increases and CO_2 phase state change become liquid to supercritical

The pressure reaches maximum value of 8.7 MPa
The temperature of collector outlet reaches maximum value of 63.3 °C

After 16:00 when the sun start setting, the temperatures and pressures goes down with changing the phase into the initial state of gas-liquid two-phase

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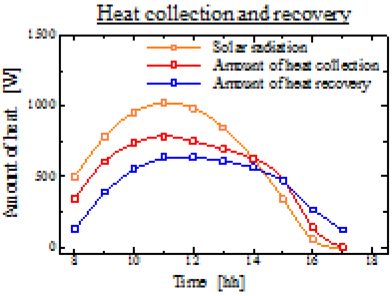
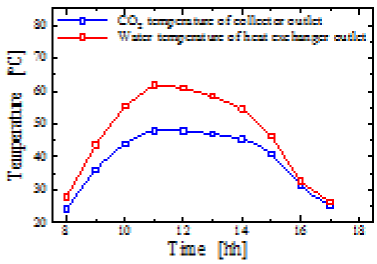
Discussion

30

Heat collection and recovery



Temperature



Solar heat recovery system

As soon as the experiment starts, the collected and the recovered heat quantities are zero
After CO_2 is driven, both of the quantities rapidly increase

	Heat collection efficiency [%]	Heat recovery efficiency [%]	Maximum temperature [°C]	
			CO_2	Water
S-CO_2	83.9	69.4	63.3	48.9
Flat plate		65.6		54.6

Flat plate collector

- Heat collection efficiency and heat recovery efficiency of S-CO_2 collector are higher than flat plate collector
- Maximum water temperature of S-CO_2 collector is lower than flat plate collector, however CO_2 temperature is higher than water temperature in flat plate collector

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Conclusions

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Heat pump system

Using the state an ultra low temperature refrigeration could be possible in a closed heat pump cycle. The development can achieve further ultra-low temperature system, a swirl promoter is newly installed at inlet of the evaporator/sublimator, by can the results be obtained;

- The dry-ice particles flow into downstream without sedimentation near inlet.
- The heat transfer is increased.
- The dry-ice particles cover with the inner pipe compared with in the case of without swirl promoter, dry-ice particles are formed sedimentation like snow ball and it flows into downstream with rotation.

Solar water heater

The cross pseudo-critical cycle can be achieved by natural circulation of CO_2 for absorption of solar thermal energy. A state of art technology has been proposed and designed as a new type of solar water heater. Some representative performance results are presented together with the results concluded;

- The natural circulation can be achieved, and circulate CO_2 in the system without using mechanical feed pump.
- Heat recovery efficiency of the system is found to be higher than ordinary flat plate collector.

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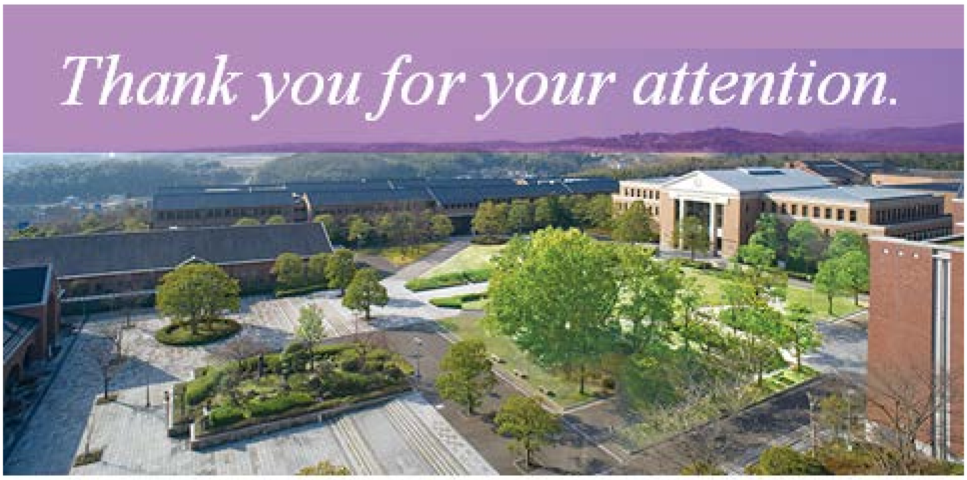

同志社大学京田辺キャンパス

Doshisha University, Kyotanabe campus, Kyoto, Japan

Doshisha University