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Abstract

Dry ice sublimation is a unique phase change phenomenon which occurs under -56 ˚C by absorbing a great deal of heat quality under triple point temperature of -56 ˚C. With applying the dry ice sublimation, CO_2 refrigeration system can be achieved in a temperature lower than -56 ˚C. In the evaporation process of actual refrigeration system, however, dry-ice blockage may be happen in an evaporator, which causes a risk of system failure associated with vacuum pressure in a suction of a compressor. In this study, a tapered evaporator/sublimator with a swirl promoter, which induced swirling flow of solid (dry ice)-gas two-phase flow, is newly designed and constructed, and actually installed into a proposed CO_2 ultra-low temperature cascade refrigeration system. By means of the heat transfer of solid (dry ice) -gas two-phase flow is investigated. Based on the measurement for heat transfer characteristics, it was verified that the CO_2 refrigeration system can operate continuously and stably without dry ice blockage in the evaporator/sublimator. It was understood that dry ice particles are uniformly distributed along the inner wall of the evaporator/sublimator by installing the swirl promoter, where the heat transfer coefficient is largely improved.

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1 Paper

Experimental Investigation of an Effect of Dry Ice Solid-Gas Two-Phase Flow in CO_2 Refrigeration System

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Abstract:

Dry ice sublimation is a unique phase change phenomenon which occurs under -56 °C by absorbing a great deal of heat quality under triple point temperature of -56 °C. With applying the dry ice sublimation, CO_2 refrigeration system can be achieved in a temperature lower than -56 °C. In the evaporation process of actual refrigeration system, however, dry-ice blockage may be happen in an evaporator, which causes a risk of system failure associated with vacuum pressure in a suction of a compressor. In this study, a tapered evaporator/sublimator with a swirl promoter, which induced swirling flow of solid (dry ice)-gas two-phase flow, is newly designed and constructed, and actually installed into a proposed CO_2 ultra-low temperature cascade refrigeration system. By means of the heat transfer of solid (dry ice) -gas two-phase flow is investigated. Based on the measurement for heat transfer characteristics, it was verified that the CO_2 refrigeration system can operate continuously and stably without dry ice blockage in the evaporator/sublimator. It was understood that dry ice particles are uniformly distributed along the inner wall of the evaporator/sublimator by installing the swirl promoter, where the heat transfer coefficient is largely improved.

Introduction:

The usage of carbon dioxide (R-744) as a refrigerant of heat pump system has been increased substantially, which the green gas property is defined as datum by 0 and 1 of ODP and GWP, respectability. CO_2 is also classified as non-flammable, non-toxic, chemically inactive, and inexpensive as well. Also, CO_2 has great potential of heat transfer owing to the volumetric capacity of CO_2 being 3 or 4 times higher than for other refrigerants available in the market. Based on the advantages owing to the reasons, CO_2 has received much attention in recent years in developing

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energy conversion systems (Aroz et al., 2008; Austin and Sumathy, 2011; Cechinato et al., 2005; Chen and Zhang, 2014; Chen and Cu, 2005; Girotto et al., 2004; Huang et al., 2007; Kim et al., 2004; Kim et al., 2009; Liao and Zhao, 2002; Liu et al., 2012; Lorentzen, 1994; Nekså et al., 1998; Rieberer, 2005; Sarkar et al., 2006; Stene, 2005; Tamura et al., 2005; White et al., 2002). For recent years, many researchers have also carried out scientific investigations for CO_2 heat pump system (Boccardi et al., 2017; Zhu et al., 2017; Yokoyama et al., 2018). To date, in the heat pump system using CO_2 as working fluid, the cooling temperature range is usually from -30.0 ~ 0 °C achieved by the evaporation process since so far developed heat pump systems focus on mainly water heater with high efficiency by transcritical CO_2 operation. For the high technology industrial applications, however, fishing industry, biomedical engineering, food industry and etc., below -30.0 °C cooling technology is much required.

Based on above advantage and moreover to overcome technical issues, a CO_2 ultra-low temperature cascade refrigeration system has been proposed (Yamaguchi et al., 2008; Yamaguchi et al., 2009). The refrigeration system can achieve a low temperature below CO_2 triple point temperature -56.6 °C at 0.518 MPa by sufficiently expanding cooled liquid CO_2 into the solid-gas two-phase flow, i.e. to the region of the dry-ice formation. The refrigeration system composed of two CO_2 refrigeration compression cycles, namely high pressure cycle (HPC) and low pressure cycle (LPC), each of which composed of condensers, an expansion valve, an evaporator and a compressor. The two CO_2 refrigeration compression cycles are arranged in cascade through a brine heat exchanger channel in order to cool liquid CO_2 in LPC for forming of dry-ice in the evaporator/sublimator. A feasibility study of the system operation using CO_2 solid-gas two-phase flow has been performed (Zhang et al., 2011), and the system has performed the capability of achieving ultra-low temperature range of -60 ~ -62 °C at the suitable experimental condition. On the contrary, when the system operates at lower temperature conditions (low condensation temperature and low heating power input) in the evaporate process, it has been found that dry-ice blockage occurs in the evaporator/sublimator and makes the system operation failing (Yamaguchi et al., 2011). The dry-ice blockage is caused that accumulated dry ice particles (or dry-ice sedimentation) block the cross section of the evaporator/sublimator. Due to the dry ice blockage, the vacuum operation of compressor in suction side follows and causes the system failure in LPC. In order to examine the sedimentation of dry-ice phenomena, a dry-ice observation system with visualization channels modelling the evaporator/sublimator has been designed, constructed and tested (Yamaguchi et al., 2014). The dry-ice visualization test has qualitatively revealed that the geometric configuration of the inlet shape of the evaporator/sublimator strongly influences the flow behavior of the dry ice solid-gas two-phase flow inducing the sedimentation phenomena. Furthermore, it revealed that the sedimentation phenomenon is greatly eased when the shape of the inlet of evaporator/sublimator is changed from a sudden expansion channel to a tapered channel. By using the shape of the inlet of evaporator/sublimator in the CO_2 cascade refrigeration system, the system has performed the capability of achieving the ultra-low temperature of -66.3 °C, continuously (Iwamoto et al., 2015).

Based on the previously reported study (Yamasaki et al., 2017), in order to improve heat transfer characteristics of CO_2 solid-gas two-phase flow in the evaporator/sublimator, a tapered channel with a swirl promoter as an inlet channel of the evaporator/sublimator is modified, in which the swirling flow of CO_2 solid-gas two-phase flow is induced. Beside the actual CO_2 heat pump system, in the present study, the dry-ice behavior is investigated by the observation system with a new visualization channel (with a swirl promoter). By inducing the swirling flow of CO_2 solid-gas two-phase in the evaporator/sublimator, it is found that heat transfer of CO_2 solid-gas two-phase flow increases due to uniformly dispersed the dry-ice particles in the whole region of the evaporator/sublimator. Based on this research the evaporator/sublimator with a swirl promoter is newly designed and constructed, and actually installed into the existed CO_2 ultra-low temperature cascade refrigeration system gives high heat transfer characteristics, improving the system performance.

Experiment:

The swirl promoter is designed and manufactured in order to improve the solid-gas two-phase heat transfer in the evaporator/sublimator in LPC of the CO_2 ultra-low temperature cascade refrigeration system as shown in Fig. 1. LPC mainly composed of three condensers, an expansion valve, an evaporator/sublimator (test section) and a compressor. In order to obtain CO_2 solid-gas two-phase flow in the evaporator/sublimator, it is necessary to condense gas CO_2 into liquid CO_2 before entering the expansion valve. Three condensers arranged in series are designed to condense gas CO_2 by sufficiently cooling in stages. The first and second condensers are tube-in-tube heat exchangers. The first condenser is cooled by hot water and the second condenser is cooled by cold water from cooling tower. The third condenser is a plate-type heat exchanger which is cooled by the brine connected with the evaporator of HPC. The expansion valve is a needle valve which is manually operated to give LPC at given condition. The compressor is a reciprocating compressor (TCS350/4, produced by Dorin).

Fig. 2 depicts the detailed configuration diagram of the newly installed tapered evaporator/sublimator with a swirl promoter in the LPC. The evaporator/sublimator is mainly composed of a tapered channel, heater, thermal insulation and swirl promoter. The swirl promoter is stainless thin wire (of 1 mm diameter) boned along the inner wall of the tapered channel as depicted in Fig. 3. The tapered channel is a copper-made horizontal placed circular pipe with the length of 5000 mm, the inner and outer diameters of 40 mm and 45 mm, respectively. The inlet of the circular pipe is tapered from inlet of the pipe x=0 mm to x=200 mm. The heater used to heat the pipe is a silicon gum type heater, which is twisted around the pipe. The heater input is controlled by an inverter deice. A thermal insulation, which has a thickness of 150 mm, made of glass wool is converted outside the pipe.

As for the measurement system, as shown in Fig. 1, thermocouples are installed between the compressor, first condenser, second condenser, third condenser, expansion valve and evaporator/sublimator, and pressure transmitters are installed between the compressor and expansion valve in the LPC to measure temperatures and pressures, with an accuracy of $\pm 0.1\%$ for temperature measurements and $\pm 0.2\%$ for pressure measurements. In order to investigate the heat transfer characteristics of solid-gas two-phase flow in the evaporator/sublimator of the LPC, four pressure measuring point (denoted as $P_1 - P_4$ in Fig. 2) are arranged with uniformly distributed along the upper wall of the evaporator/sublimator, and 15 thermocouples (denoted as $T_1 - T_{15}$ in Fig. 2) are positioned to measure wall temperatures of the circular pipe.

In the present study, in order to investigate the heat transfer of solid-gas two-phase flow in the evaporator/sublimator of LPC, the condensation temperature is set to be -20 °C by controlling the heat pump system of HPC. It is noted that there are enough reference data of heat transfer in evaporator/sublimator at the condensation temperature of -20 °C (Yamaguchi et al., 2009). As with the expansion valve, the opening of 25 mm is set when the opening ratio is 5/6. The heater input is 1800 W (2904 W/m^2). The rotation speed of the compressor is set at 55 Hz controlled and throughout the experiments.

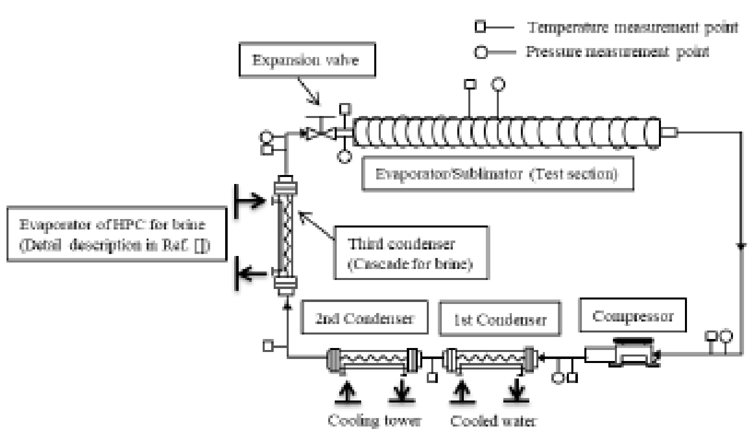


Fig. 1 Arrangement of low pressure cycle (LPC) in experimental CO_2 ultra-low temperature cascade refrigeration

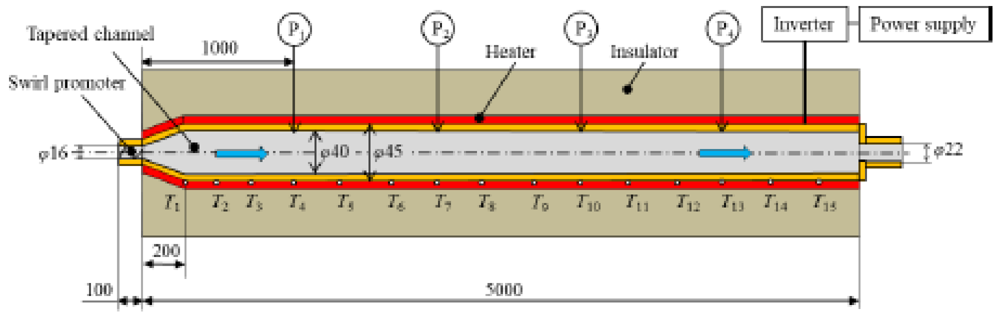


Fig. 2 Assembly of tapered evaporator/sublimator with the swirl promoter

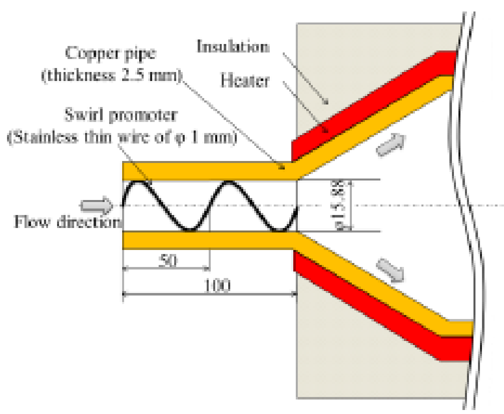


Fig. 3 Detail view of the inlet flow channel

Results and discussion:

In order to prevent the dry-ice blockage and increase the heat transfer, in this study, the swirl promoter is actually installed as explained above in the CO_2 ultra-low temperature cascade heat pump system, and the system operation and the heat transfer are tested. System operation without a swirl promoter (only tapered flow channel) is also tested for comparison.

Fig. 4 shows the measured pressure both at discharge and suction compressor with the swirl promoter installed. In the Fig. 4, the lateral axis shows the time variation. It is mentioned here that at the beginning of experiments, the heat pump system as shown in Fig. 1 is operated without the heater input to the evaporator/sublimator in LPC in order to obtain sufficiently cooled CO_2 before the expansion valve, and when the brine is fully cooled, the heat input to the evaporator/sublimator and this time point is set as the point of origin of the experiment as shown time=0 min. As shown in Fig. 4, the system the steady state after 10 min, and the time-average valves of the suction pressure is taken as 0.44 ± 0.02 MPa and the discharge pressure is taken as 2.04 ± 0.07 MPa. It is confirmed that the pressure of evaporator/sublimator is below the CO_2 triple point of 0.518 MPa. It should be noted here that the oscillations of both of discharge and suction pressures are quite low compared with previous study (Yamaguchi et al., 2011) which used only sudden expansion channel (without the swirl promoter) as the inlet shape of the evaporator/sublimator. It is verified that the CO_2 refrigeration system with the swirl promoter could produce more stable operation than the previous system. Fig. 5 shows average wall temperatures in the range of 0~2000 mm (T_1~T_7) and 2000~5000 mm (T_8~T_{15}) of the evaporator/sublimator with the swirl promoter in LPC. Similarly after 10 min, in the range of 0~2000 mm (T_1~T_7) of the evaporator/sublimator in LPC, the average wall temperature is -39.1 ±0.25°C for 140 min indicating that the sublimation rate of dry ice is balanced with the generation rate of dry ice. In the range of 2000~5000 mm (T_8~T_{15}) of the evaporator/sublimator in LPC, the average wall temperature is -30.2 ±0.73 °C after 10 min. However, there is some intermittency of temperature variations, in which the highest peak is approximately -26 °C. This is due to local pile up (or aggregation of dry ice particle) with which the smooth flow may have disturbed. The average wall temperature in after 2000 mm of the evaporator/sublimator becomes 9 °C higher than the temperature before 2000 mm of the evaporator/sublimator. It is considered that the rate of sublimation decreases, when local dry ice sedimentation is occurred. The local dry ice sedimentation may cause the accumulated dry ice particles on the bottom wall of take, preventing active sublimation.

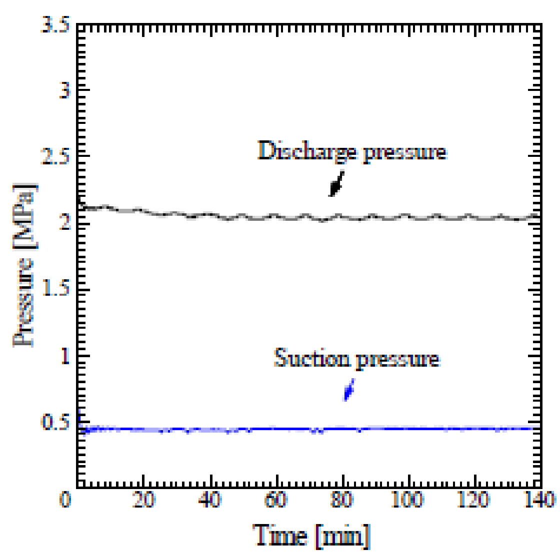


Fig. 4 Variations of pressures with time with swirl promoter

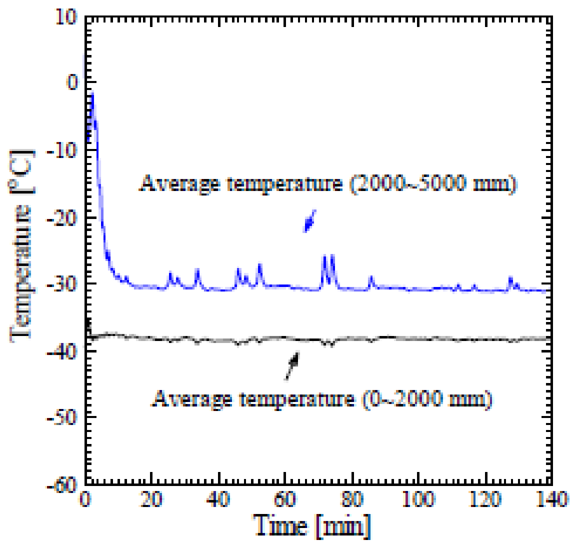


Fig. 5 Variations of measured evaporating temperatures of LPC with time at tapered channel with the swirl promoter.

Here, in order to show the advantage of using the swirl promoter in the evaporator/sublimator, results of measurement without the swirl promoter is compared in Fig. 6 as a reference, where measurements of evaporating pressure and average wall temperatures in the range of 0~2000 mm (T_1~T_7) and 2000~5000 mm (T_8~T_{15}) of the evaporator/sublimator as displayed in the same experimental condition as Fig. 5. As shown in Fig. 6, the suction pressure rapidly decreases during the time between 18 and 38 min. The change of the suction pressure is caused by a large amount of the dry ice accumulation at the inlet small tube before the compressor (Fig. 2), blocking the gas CO_2 flow. Average wall temperatures also dramatically change while suction pressure change. Focusing on the result of average temperature in the range of x=0 - 2000 mm of the evaporator/sublimator, the average temperature decreases within 19 min caused by increasing dry ice precipitation from eases generation of dry ice, and then temperature rises up. When dry ice precipitates to such an extent that sublimation rate decreases and sedimentation of dry ice grows and eventually causes a blocking the evaporator/sublimator. Since majority of dry ice particles sublimated at the inlet part of the evaporator/sublimator, gaseous phase with large agglomeration of dry ice flow into the end part of the evaporator/sublimator. This phenomena would cause the average wall temperature dramatically increases during the time range of 19~24 min.

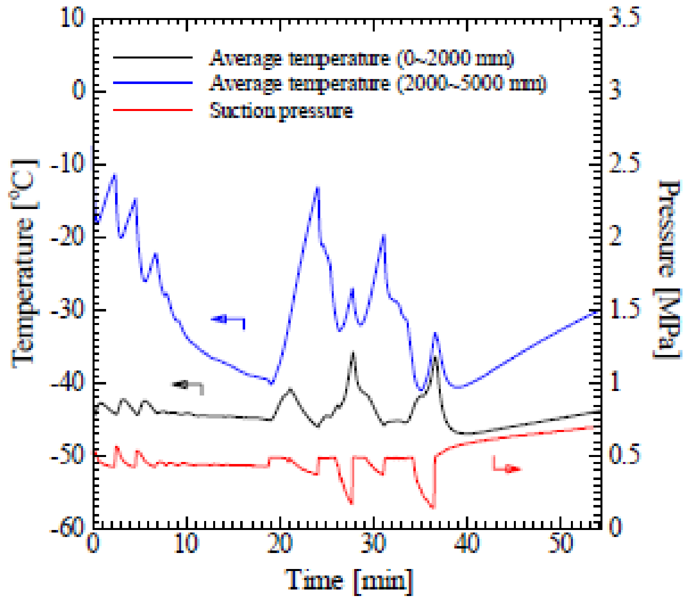
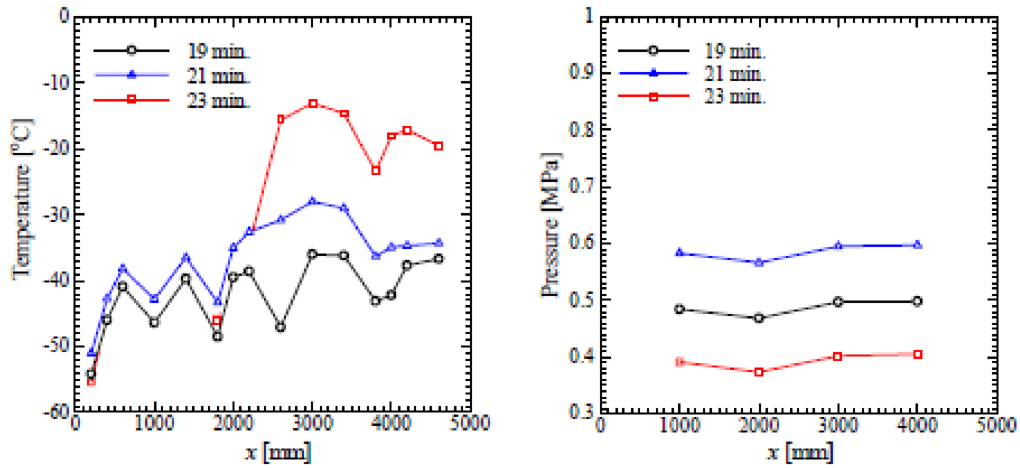


Fig. 6 Variations of measured evaporating pressure and temperatures for only tapered channel without swirl promoter

In order to investigate the detail heat transfer of solid-gas two-phase flow, the variations of local pressures and temperatures at elapsing time of 19, 21 and 23 min (typical blocking time progress) in case of using the tapered channel without the swirl promoter are shown in Fig. 7. At the time of 19 min, as seen the local temperatures slightly increase with oscillation into the downward direction. At this instance the solid two-phase flow does not distribute uniformly in the pipe, although dry ice particles distribute with a certain amount of sedimentation at the bottom of the pipe, so that the measured temperature tends to oscillate (Yamasaki et al., 2017). At the downstream, since the amount of sublimation of dry ice decreases, the heat transfer decreases with temperatures slightly increasing. At the time of 21 min, in the range of *x*=2000~3800 mm, it is similarly seen that the temperature also increases. This is caused that strong large dry ice agglomeration is formed at bottom wall in this range. When the large agglomeration of dry ice is formed in the evaporator/sublimator, heat transfer is degraded as major amount of heat is transported by heat conduction. Moreover, it is considered that the large agglomeration blocks the evaporator/sublimator, because of the pressure exceeds the sublimation pressure (0.518 MPa) of CO_2 as seen in Fig. 7(b). Once the dry ice blockage eliminated from the evaporator/sublimator, the pressures decrease owing to the fact of Joule-Thomson effect taking place as seen in Fig, 7(b) at the time of 23 min. After the blockage phenomena occurs, however, since the discharge amount of CO_2 decreases, there is not enough precipitation of dry ice, and solid-gas two-phase flow changes to single gaseous phase after 2000 mm, which leads that the local heat transfer coefficient decreases as shown in Fig. 7(a) at the time of 23 min.



(a) local temperature distributions (b) local pressure distributions

Fig. 7 Variations of measured (a) local temperatures and (b) local pressure at 19, 21, 23 min using tapered channel without swirl promoter

Fig. 8 shows the variations of measured temperature and pressure at 40, 80 and 120 min in the case of using the swirl promoter. Measurement results of various local temperatures and pressures in the evaporator/sublimator at whole operation time (140 min) shows almost the same value. As mentioned above in Fig. 4 and 5, by installing the swirl promoter in the inlet channel of the evaporator/sublimator, the system can be operated continuously and stably without dry ice blockage in the evaporator/sublimator. As clearly shown in Fig. 8, the temperature increases from 0 to 2800 mm of the evaporator/sublimator with some degree of oscillation and after 2800 mm local point it decreases, reaching -35 °C at 3000 mm. It is interesting to note that the temperature keeps almost constant value of -35 °C until the local point of 4200 mm and at the end of the evaporator/sublimator, temperature slightly increases. Here it is thought that the same amount of dry ice can be induced to the evaporator/sublimator as observed from pressures taking almost the same value which is lower than CO_2 sublimation pressure of 0.518 MPa. The increasing trend of temperatures in the range of 0 – 2800 mm is well known effect of developing the thermal boundary layer, indicating that the temperature difference between the inner wall temperature and the CO_2 temperature becomes larger along the evaporator/sublimator. The cause of large oscillation of the temperatures in the range of 0 – 2800 mm can be considered as the spiral trajectory of the dry ice swirling flow. Especially in the entrance region of the evaporator/sublimator (0~1400 mm), the dry ice particles in the solid-gas two-phase flow induced by the swirl promoter flow prevails strong spiral trajectory motion. This trend is conceivable from the previous visualization results (Yamasaki et al., 2017). After x=3000 mm, the constant temperature is maintained by solid-gas two-phase flow distribution throughout the pipe. After the sublimation, the temperature at x=4600 mm increases due to the dry ice passing into the single gaseous phase.

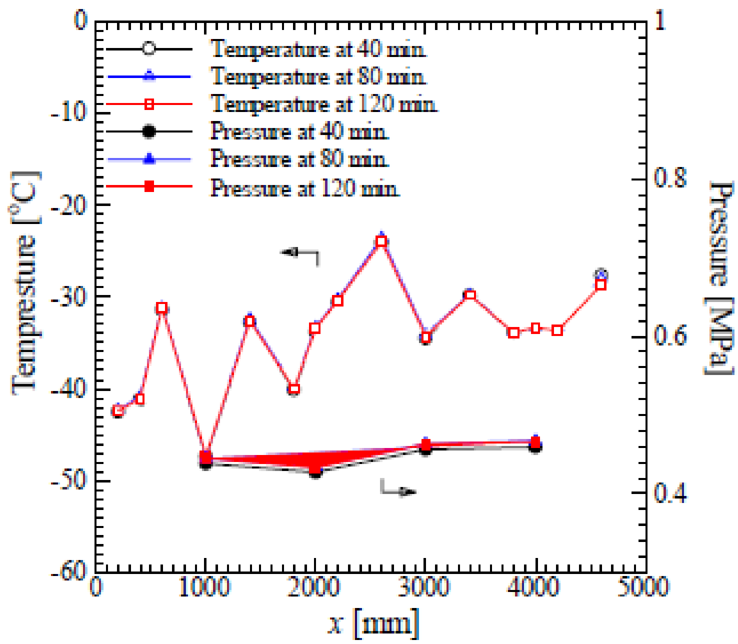


Fig. 8 Variations of measured temperatures and pressures at 40, 80 and 120 min. in case of using the swirl promoter.

In order to discuss the heat transfer coefficient with and without the swirl promoter, the heat transfer characteristics of the solid-gas two-phase flow inside of the evaporator/sublimator is studied in details as shown in Fig. 9. In the figure, the lateral axis shows the position x from the inlet of the evaporator/sublimator and the longitudinal axis represents the local heat transfer coefficient h_x as calculated by the following formula;

$$h_x = \frac{q}{T_w - T_{in}}$$

where q is the input heat flux, T_w is the inner wall temperature of the pipe and T_{in} is the CO_2 temperature in the evaporator/sublimator. The CO_2 temperature in the evaporator/sublimator T_{in} is calculated from the measured pressure at the same position by a Program Package for Thermo-physical Properties of Fluids database 12.1 (PROPATH v12.1) (PROPATH GROUP, 2001) and NIST Reference Fluid Thermodynamics and Transport Properties (REFPROP 8.0). As shown in Fig. 9, the local heat transfer coefficient at x=1000 mm in the case of with swirl promoter is higher than that in the case of without swirl promoter. This is caused by inducing a swirling flow that a large amount of dry ice particle may be dispersing along the inner wall of the pipe by absorbing great deal of heat quality. In the range of x=1000 – 2000 mm, both of the heat transfer coefficient show decrease, indicating development of thermal boundary layer of gaseous phase along the wall. In the case with of swirl promoter, it is understood that the heat transfer coefficient is increased by active sublimation heat transport by dry ice. In the case of without swirl promoter, the heat transfer coefficient tends to decreasing at x=3000 mm and then increasing at x=4000 mm. These trends can be explained by the large agglomeration of dry ice being formed in the evaporator/sublimator, where the heat transfer is degraded as the major heat is transported heat conduction. Furthermore, when the sedimentation of dry ice that fills the bottom wall of the pipe occurs, it is considered that at this condition vapor layer is formed between the inner wall and the sedimentation of dry ice, leading the heat transfer coefficient further deteriorating. At x=4000 mm as shown in Fig. 9 in the case of with swirl promoter, the solid-gas two-phase changes into the single gaseous phase, which leads that the local heat transfer coefficient decreases along the evaporator length, owing to thick thermal boundary developing, and the heat transfer coefficient increase. On the other hand, in the case of without the swirl promoter, the heat transfer coefficient increases at x=4000. This is caused of increase in the heat transfer coefficient because the residual dry ice sedimentation flows from upwind.

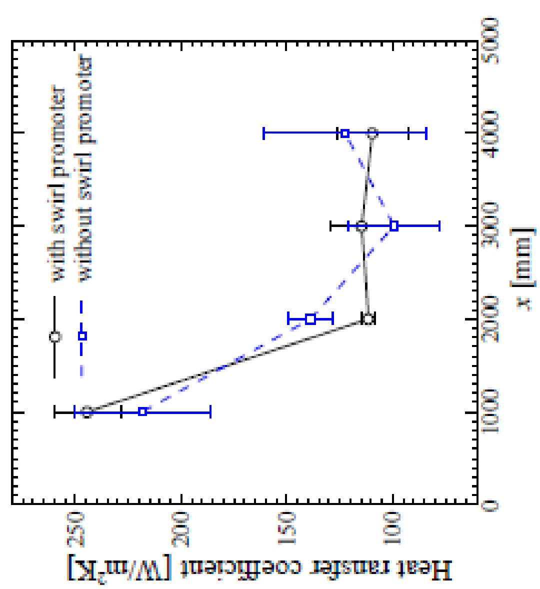


Fig. 9 Heat transfer coefficient in case with and without the swirl promoter

Conclusion:

In the present work, the tapered evaporator/sublimator with a swirl promoter is newly designed and constructed, and actually installed into the proposed CO_2 ultra-low temperature CO_2 cascade refrigeration system. In order to improve the heat transfer of solid-gas two-phase flow, the heat transfer characteristics of CO_2 solid (dry ice)-gas two-phase flow in the evaporator/sublimator are investigated.

By installing the swirl promoter in the evaporator/sublimator, it is found that the system can be operated stably and continuously owing to well developing the thermal boundary layer with dry ice sublimation. And dry ice particles are uniformly distributed along the inner wall of the evaporator/sublimator by installing the swirl promoter, where the heat transfer coefficient is largely improved.

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