Experimental Investigation of the Effect of Solid-Gas Two-Phase Flow in CO2 Cascade Refrigeration System

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Abstract

The dry ice sublimation process of CO_2 is a unique technique in which temperature ranges below the triple point of -56 °C can be achieved in a CO_2 refrigeration system. However, during the evaporation process of the actual refrigeration system, the dry-ice blockage maybe happens in the evaporator, which causes a risk of system failure in the suction of the compressor. In order to overcome this problem, in this study, an ultra-low temperature CO_2 cascade refrigeration system with a novel tapered evaporator/sublimator was designed and constructed. The novel evaporator/sublimator included a swirl promoter, which induces the swirling flow of solid-gas two-phase flow. Experiments were conducted for the investigation of solid-gas two-phase flow heat transfer characteristics in the evaporator/sublimator. According to the experimental results, it is verified that the CO_2 refrigeration system can operate consistently and steadily without dry ice blockage in the evaporator/sublimator. In addition, the dry ice particles are uniformly distributed along the inner wall of the evaporator/sublimator by the installation of the swirl promoter, and

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the heat transfer coefficient is considerably improved.

1. Introduction

Recently, the utilization of carbon dioxide (CO_2) as a refrigerant for the refrigeration systems has been raised significantly, in which the green gas property is defined as datum by 0 and 1 of Ozone Depletion Potential (ODP) and Global Warming Potential (GWP), respectively [1-2]. CO_2 is also classified as non-flammable, non-toxic, chemically inactive, and inexpensive as well [3]. Furthermore, CO_2 has excellent potential for heat transfer characteristics owing to the volumetric capacity, which is 3 or 4 times higher than other refrigerants available in the market [4]. Based on these advantages, CO_2 has attracted more attention in the field of energy conversion technologies and refrigeration systems [5].

In recent years, a number of researchers have carried out scientific investigations for CO_2 refrigeration systems [6]. Many of them are related to performance improvement and new design configurations. A comprehensive review study on the latest developments of CO_2 refrigeration was done by Yu et al. [7]. Their review consisted of the latest studies on the advanced transcritical CO_2 refrigeration technologies, especially for the last two decades, and according to their summary, besides their some drawbacks, these technologies were reported to be promising for the refrigeration applications. Another review study was performed by Dilshad et al. [8], focusing on the recent progress of CO_2 based HVAC systems. According to these review studies, several methods and different cycle configurations are reported in the literature for the CO_2 refrigeration. Llopis et al. [9] conducted performance analyses of the different types of cascade refrigeration cycles. They analyzed the systems

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for different configurations using CO_2 or different refrigerant combinations in the lower cycle. They concluded that among the environmentally friendly refrigerants, CO_2 was the most promising refrigerant for the lower cycle, especially for warm regions. In their other study, they performed a literature survey on the subcooling methods of CO_2 refrigeration cycles. The subcooling method was considered as a path for performance improvement of the system [10]. Torrella et al. [11] experimentally investigated the characteristics of an internal heat exchanger in a transcritical CO_2 refrigeration system. They conducted the analyses by driving the system with and without internal heat exchanger for different evaporator and gas cooler temperatures. They reported that with the utilization of the heat exchanger, system efficiency and refrigeration capacity were both increased. In addition to these studies, further developments of the CO_2 refrigeration cycles have been investigated by researchers such as performance of a cooling system with different cascade configurations using CO_2/NH_3 [12], the effect of expansion valve opening on the CO_2 based refrigeration performance [13], effect of thermoelectric module integrated subcooling system on the transcritical CO_2 cooling system [14], implementing a booster system with parallel compression [15], integration of CO_2 refrigeration system with two-stage parallel compression and solar absorption refrigeration system [16], partial cascaded two-stage CO_2 refrigeration system [17], CO_2 transcritical refrigeration system with ejector expansion and two-stage evaporation [18] and CO_2 two-stage cooling system integrated with two ejectors [19].

To date, for the CO_2 refrigeration systems, the cooling temperature generally varies between -30.0 ~ 0 ℃ acquired by the evaporation method. On the other hand, for the high technology industrial applications such as the fishing industry, biomedical engineering, food industry, etc., cooling technology below -30.0 ℃ is much required [20]. Besides, the range of proper refrigerants for the low-temperature refrigeration is limited due to the

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low evaporation temperature necessity, which should be higher than the boiling point of the working fluid. In a comprehensive review study which was carried out by Babikoni et al. [21] on the ultra low-temperature refrigeration systems, it was reported that there are some limitations for the possible refrigerants such as higher GWP values (R23 and R508B), flammability (R170 and R1150) and limited available information (R1132a). Besides, a substantial number of related literature depend on theoretical studies. Furthermore, in a review study of Bansal [22] on the CO_2 based low-temperature refrigeration systems, CO_2 is reported as the most promising refrigerant for ultra-low refrigeration systems for the temperature ranges between -25 °C ~ -50 °C due to its favorable thermophysical characteristics. Huang et al. [23] theoretically investigated a new CO_2 refrigeration system using vapor and solid CO_2. Due to the triple temperature of the CO_2, which is -56 °C, they used a nozzle, a sublimator, high, and low pressure regulating valves instead of the evaporator. According to their results, they concluded that the system could achieve working temperatures of the below triple point of CO_2.

Based on the above-mentioned advantages of CO_2 and to overcome technical limitations reported in the previous studies by many authors, a novel experimental cascade refrigeration system working with CO_2 has been proposed and constructed by Yamaguchi et al. [24] for ultra-low temperature applications. The experimental system consisted of two refrigeration cycles, where CO_2 was employed as a working fluid in the high-pressure cycle (HPC) and low-pressure cycle (LPC). The two cycles were designed in cascade arrangement through a brine heat exchanger channel in order to cool liquid CO_2 in LPC for the dry-ice formation in the evaporator/sublimator. It was aimed by the experimental study to achieve low-temperature refrigeration below the triple point temperature of CO_2, which is -56.6 °C at 0.518 MPa by sufficient expansion of liquid CO_2 into the solid-gas two-phase state. After a series of experiments, the system has shown promising performance

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in achieving ultra-low temperature ranges of -60 ~ -62 °C by CO_2 solid-gas two-phase flow [25-26]. On the contrary, during the experiments performed for lower condensation temperature and lower refrigeration power conditions, dry-ice blockage formation observed in the evaporator/sublimator, which resulted in system operation failure. It was reported that the dry-ice blockage formation was due to the accumulated dry ice particles (or dry-ice sedimentation), which blocks the cross-section of the evaporator/sublimator and at the same time terminates the suction of the compressor in LPC [27-28]. In order to examine the sedimentation of the dry-ice phenomenon, an evaporator/sublimator with visualization channels has been designed, constructed, and experimented for the observation of the dry-ice formation in a further study. The dry-ice visualization experiments have 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 phenomenon. Furthermore, it was revealed that the sedimentation phenomenon was considerably eased when the shape of the inlet of the evaporator/sublimator was modified to the tapered channel from the sudden expansion channel. By modifying the shape of the inlet of the evaporator/sublimator, the CO_2 cascade refrigeration system has achieved the capability of continuously maintaining ultra-low refrigeration operation at a temperature of -66.3 °C [29-30].

Although a number of experimental studies have been conducted to achieve ultra-low temperature ranges as reported above, a more systematic analysis is required for improving the heat transfer characteristics of CO_2 solid-gas flow in the evaporator/sublimator. For this aim, in order to enhance the refrigeration performance, the evaporator/sublimator of the experimental system is modified by integrating a tapered channel with a swirl promoter, which enables the swirling flow of solid-gas CO_2.

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Besides the experimental cascade refrigeration system, the dry-ice behavior is investigated by the observation system with a newly integrated visualization channel (a swirl promoter). In the present paper, the results of the experiments of the modified system performed by inducing the swirling flow of CO_2 solid-gas two-phase in the evaporator/sublimator are given. Based on this research, the evaporator/sublimator with a novel swirl promoter is designed and actually installed to the CO_2 cascade refrigeration system to improve the system performance for ultra-low temperature applications.

2. Experimental Test-Rig

The CO_2 cascade refrigeration system consists of a low-pressure cycle (LPC) and a high-pressure cycle (LPC), respectively, as stated previously. In the modified setup, the swirl promoter is designed and manufactured to improve the solid-gas two-phase heat transfer characteristics in the evaporator/sublimator of LPC in the ultra-low temperature CO_2 cascade refrigeration system, as shown in Figure 1. A detailed description of HPC can be found in the previous report [25]. 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, the condensation of gas CO_2 into liquid CO_2 is necessary before entering the expansion valve. In the LPC, three condensers are arranged in series for the sufficient condensation of gas CO_2. The first and second condensers are tube-in-tube heat exchangers. In the first condenser, CO_2 is cooled to nearly 70 °C by means of water which is about 60 °C, and it is cooled to almost 40 °C in the second condenser by 20 °C of cold water maintained from the cooling tower. The third condenser, which is a plate-type heat exchanger, is refrigerated by the brine connected to the evaporator of HPC. With the help of serial arranged three condensers, the CO_2 is cooled into a liquid state. A needle-type expansion valve is utilized in LPC, which is manually operated in order to accurately adjust the experimental conditions. The compressor is a

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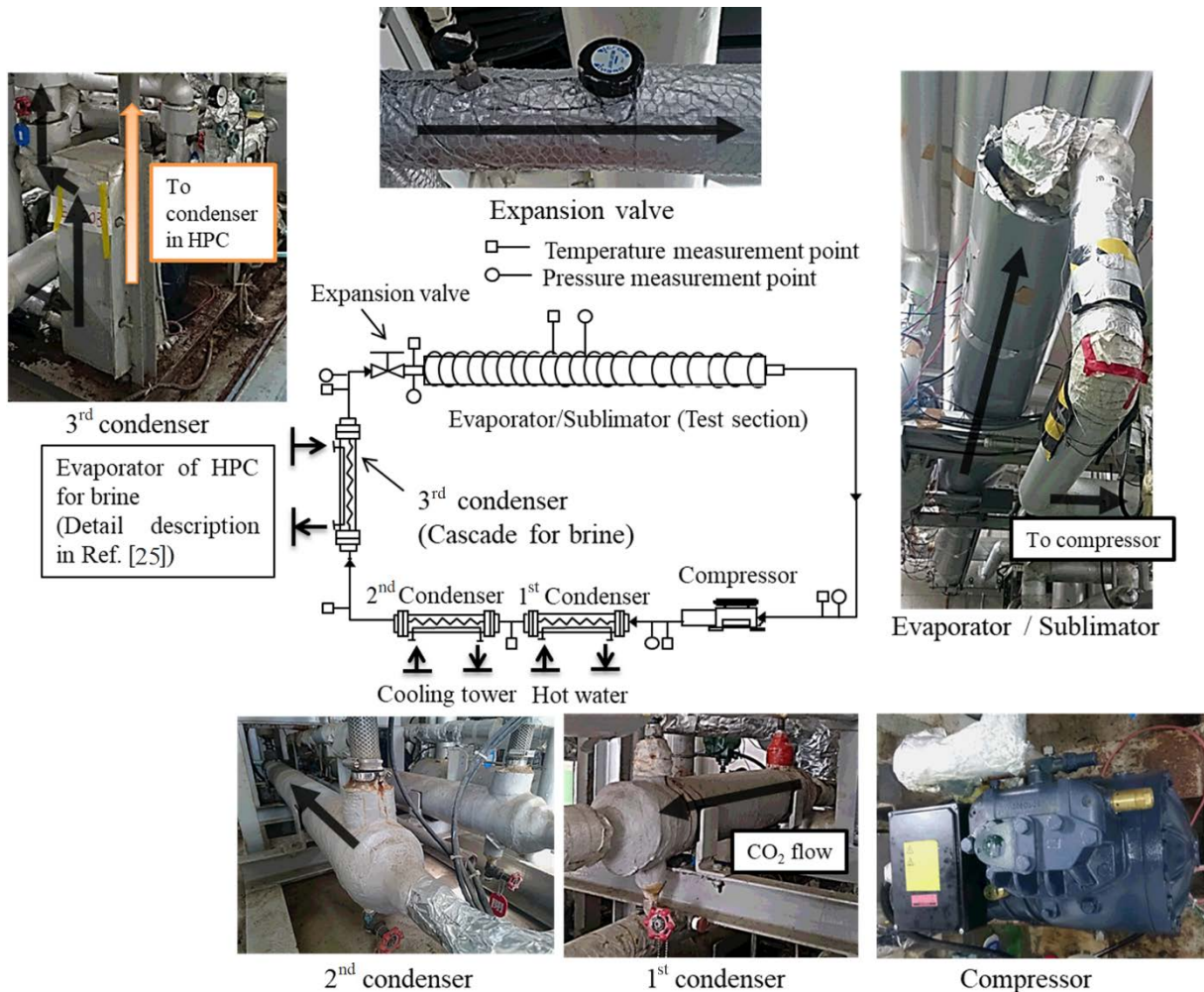reciprocating compressor (TCS350/4, produced by Dorin).

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

Figure 2 depicts the detailed assembly of the newly installed tapered evaporator/sublimator with a swirl promoter. The evaporator/sublimator is composed of a tapered channel, a heater, thermal insulation, and a swirl promoter. The swirl promoter is made of a thin stainless wire, which is 1 mm in diameter and located at the inlet of the tapered channel, as shown in Figure 3. The tapered channel is a horizontal copper-made circular pipe with a length of 5000 mm. The inner diameter of the channel is 40 mm, with

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a wall thickness of 2.5 mm. The inlet of the circular pipe is tapered-shaped, where the base diameter is 200 mm. The heater, which is twisted around the pipe, is a silicon gum type heater, and the heater capacity is controlled by an inverter device. Thermal insulation, which is 150 mm in thickness, is used to avoid heat loss during the experiments. The thermal conductivity of the insulation is 0.043 W/mK according to the manufacturer's data and made of glass wool. In addition, to prevent radiation heat loss and water vapor migration, the thermal insulation is covered with aluminum glass sheets. Thus, in the present study, the heat loss to/from the system to the environment is neglected during the experiments.
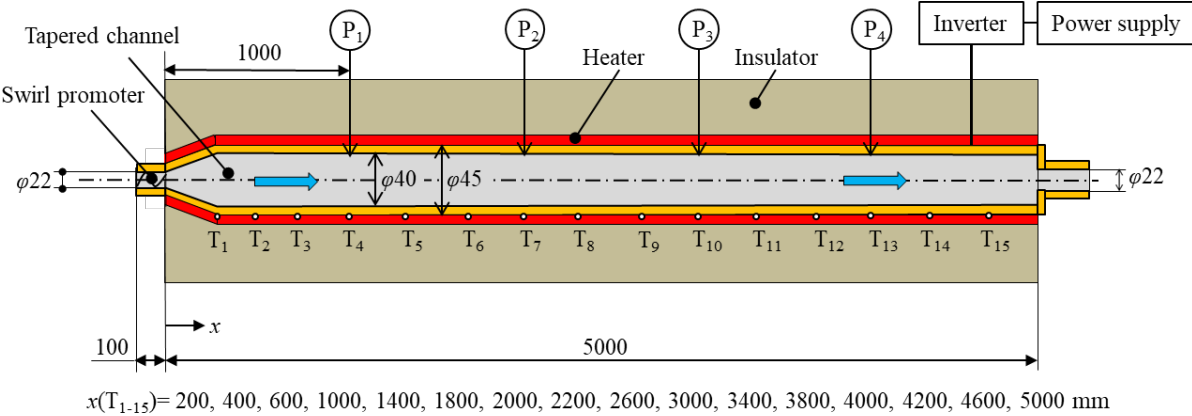

$x(T_{1-15})$= 200, 400, 600, 1000, 1400, 1800, 2000, 2200, 2600, 3000, 3400, 3800, 4000, 4200, 4600, 5000 mm

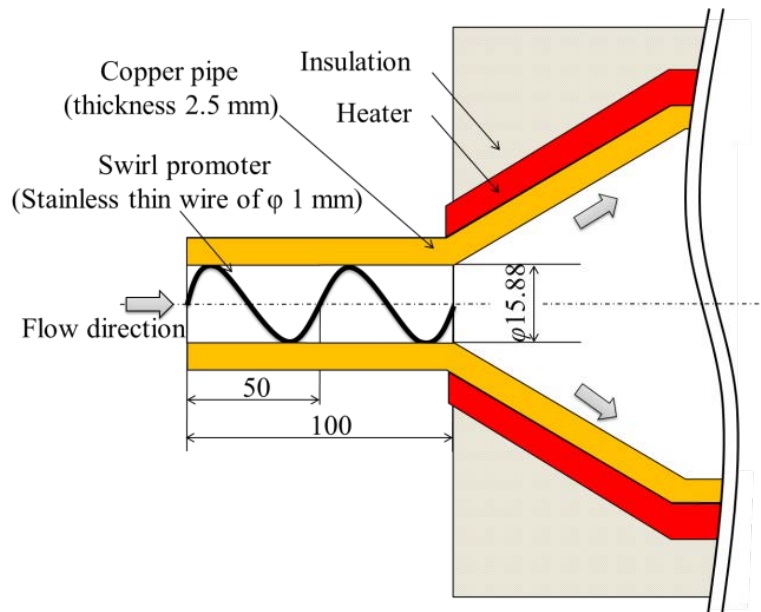Figure 2. Assembly of tapered evaporator/sublimator with the swirl promoter

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Figure 3. Detailed view of the inlet flow channel

3. Experimental Procedure

As shown in Figure 1 with squares and circles, temperature and pressure measurements are made from several different points of the system. For the temperature measurements, T-type thermocouples with an accuracy of ±0.1% are used where the accuracy of pressure transmitters is ±0.2%. In order to accurately investigate the heat transfer characteristics of solid-gas CO_2 flow, the pressure measurements are made from four different points of the evaporator/sublimator of the LPC (denoted as P_1 – P_4 in Figure 2). These pressure measurement points are uniformly distributed along the upper wall of the evaporator/sublimator. In addition, 15 thermocouples (denoted as T_1 – T_{15} in Figure 2) are positioned at the bottom wall of the evaporator/sublimator to measure the wall temperatures of the circular pipe. The measurements are recorded in every 0.2 s.

During the experiments, the condensation temperature is set to -20 °C by controlling the HPC. The opening of the expansion valve adjusted to 25 mm, and the heater input is adjusted to 1800 W (2904 W/m²) by the inverter device. For a constant rotation speed of

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the compressor, the frequency is set to 55 Hz. In a typical experiment, HTC is operated initially in order to cool the brine of the subsystem. After the brine is completely refrigerated, the heater, which is rounded over the evaporator/sublimator, is switched on for preheating. When the expansion valve of the LTC attains the predetermined temperature, the LTC begins to operate. Subsequently, the measurements are recorded during the concurrent operation of the HPC and LPC.

4. Results and discussion

In order to prevent the dry-ice blockage and increase the heat transfer characteristics, in this study, a novel swirl promoter was installed to the CO_2 ultra-low temperature cascade refrigeration system, as explained previously.   Experiments were performed, and the data was recorded for the investigation of the performance of the system operation and the heat transfer. System performance without the swirl promoter (only tapered flow channel) was also tested for comparison.

Figure 4 shows the measured pressure both at compressor discharge and suction with the installed swirl promoter. In the figure, the lateral axis shows the time variation. It is mentioned here that at the beginning of the experiments, the refrigeration system shown in Figure 1 was operated without the heat supply to the evaporator/sublimator in LPC in order to obtain sufficiently cooled CO_2 before the expansion valve. When the brine was fully cooled, the heat input was supplied to the evaporator/sublimator, and this moment was admitted as the point of origin of the experiment, i.e., time=0 min. It can be seen from Figure 4 that the system reaches the steady-state after 10 min, and the time-average value of the suction pressure is determined as 0.44±0.02 MPa where it is 2.04±0.07 MPa for the discharge pressure. It is confirmed that the pressure of the evaporator/sublimator is below the CO_2 triple point of 0.518 MPa. It should be noted here that the oscillations of

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both the discharge and suction pressures are quite low compared with the previous study [27]. In this research, only sudden expansion channels (without the swirl promoter) were used as the inlet shape of the evaporator/sublimator. It is verified that the CO_2 refrigeration system with the swirl promoter can produce a more stable operation than the previous system.
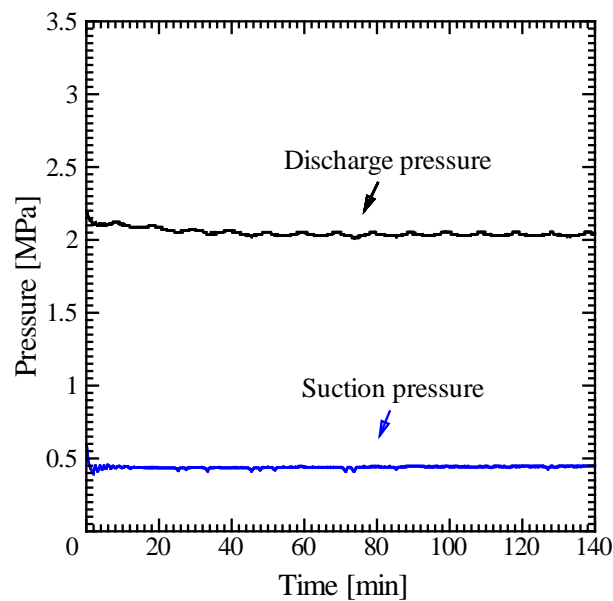

Figure 4. Variations of pressures with time for the case of swirl promoter

Figure 5 shows the average wall temperatures for 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 the 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 measured as -39.1 ±0.25°C for a timeframe of 140 min. This result indicates 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 the 10th min. However, there is some intermittency of temperature variations, in which the

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highest peak is approximately -26 °C due to the local pile up (or aggregation of dry ice particles). The average wall temperature after 2000 mm of the evaporator/sublimator becomes 9 °C higher than the temperature before. This increase in temperature can be attributed to two factors; the sedimentation of dry ice and the increase in the CO_2 gas fraction. When local dry ice sedimentation occurs, the rate of sublimation decreases. The local dry ice sedimentation may cause the accumulated dry ice particles on the bottom wall, preventing active sublimation. Moreover, near the exit of the evaporator / sublimator, the dry ice sedimentation transforms into the gas phase, which leads to a decrease in the local heat transfer coefficient owing to the thermal boundary development.
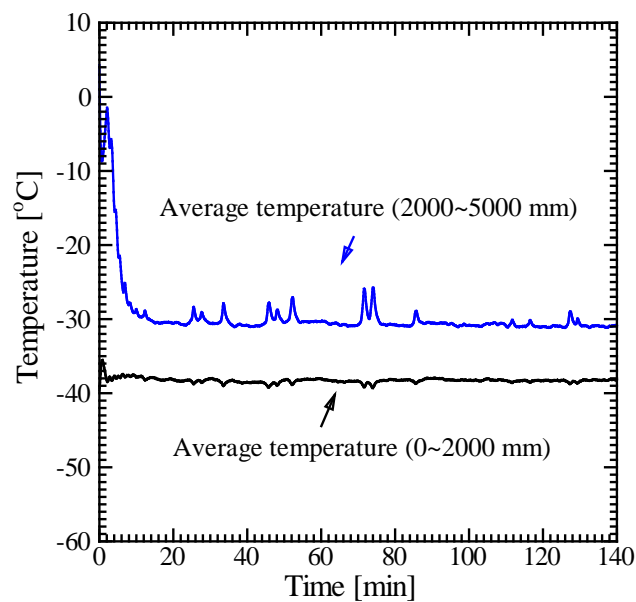

Figure 5. Variations of measured evaporating temperatures of LPC with time at the tapered channel with the swirl promoter.

In order to better understand the usefulness of the swirl promoter in the evaporator/sublimator, Figure 6 shows the results of experiments conducted without utilizing the swirl promoter in the LPC. The figure is plotted using the measurements of evaporating pressure and average wall temperatures in the range of 0~2000 mm (T_1~T_7)

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and 2000∼5000 mm (T_8∼T_{15}) of the evaporator/sublimator with the same experimental conditions as in Figure 5, for comparison purposes. As seen from the figure, the suction pressure rapidly decreases between 18 and 38 min. The sudden change in the suction pressure is caused by a large amount of the dry ice accumulation at the small inlet tube before the compressor (Figure 2), blocking the gas CO_2 flow. Average wall temperatures also dramatically change with the variation of the suction pressure. Focusing on the result of average temperature in the range of x=0 - 2000 mm, the average temperature decreases within 19 min due to the increase of the dry ice precipitation, and then the temperature starts to increase. When dry ice precipitates to such an extent, the sublimation rate decreases, and sedimentation of dry ice increases and eventually blocks the evaporator/sublimator. Due to the increased pressure upstream of the grown dry ice precipitation, the gaseous phase CO_2 flows into the ending part of the evaporator/sublimator with the large agglomeration of dry ice. This phenomenon causes a dramatic increase in the average wall temperature during the timeframe of 19∼24 min.
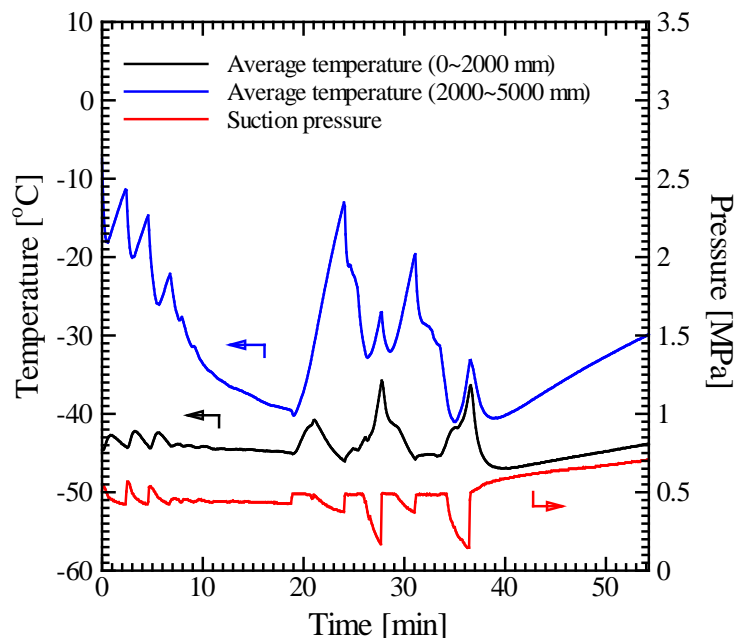

Figure 6. Variations of measured evaporating pressure and temperatures for the only tapered channel without swirl promoter

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In order to investigate the heat transfer characteristics of solid-gas two-phase flow the in detail, the variations of local pressures and temperatures at elapsed times of 19, 21 and 23 min (typical blocking time progress) are shown in Figure 7 in case of using the tapered channel without the swirl promoter. At the time of 19 min, the local temperature slightly increases with oscillation. In this instance, the solid two-phase flow does not distribute uniformly in the pipe. However, the dry ice particles distribute with a certain amount of sedimentation at the bottom where the thermo-couples installed, so that the measured temperature tends to oscillate [30]. At the downstream, since the amount of sublimation of dry ice decreases, the heat transfer rate decreases with the slightly increased temperature. At the time of 21 min, in the range of x=2000~3800 mm, similarly, it can be observed that the temperature also increases. This is due to the formation of the large dry ice agglomeration at the bottom wall. When the large agglomeration of dry ice is formed in the evaporator/sublimator, heat transfer is degraded since the major amount of heat is transported by heat conduction. Moreover, it is considered that the large agglomeration blocks the evaporator/sublimator because the pressure exceeds the sublimation pressure (0.518 MPa) of CO_2, as seen in Figure 7(b). Once the dry ice blockage eliminated from the evaporator/sublimator, the pressures decrease owing to the fact that the Joule-Thomson effect takes place, as seen in Figure 7(b) at the time of 23 min. However, after the blockage phenomenon occurs, there is not enough precipitation of dry ice since the discharge amount of CO_2 decreases, and solid-gas two-phase flow transforms to gaseous phase after 2000 mm, which leads a decrease in the local heat transfer coefficient as shown in Figure 7(a) at the time of 23 min. At the times of 19, 21, and 23 min, after x=3800 mm, it is seen that the temperature increases again. This is because the dry ice sedimentation transforms into the gas phase.
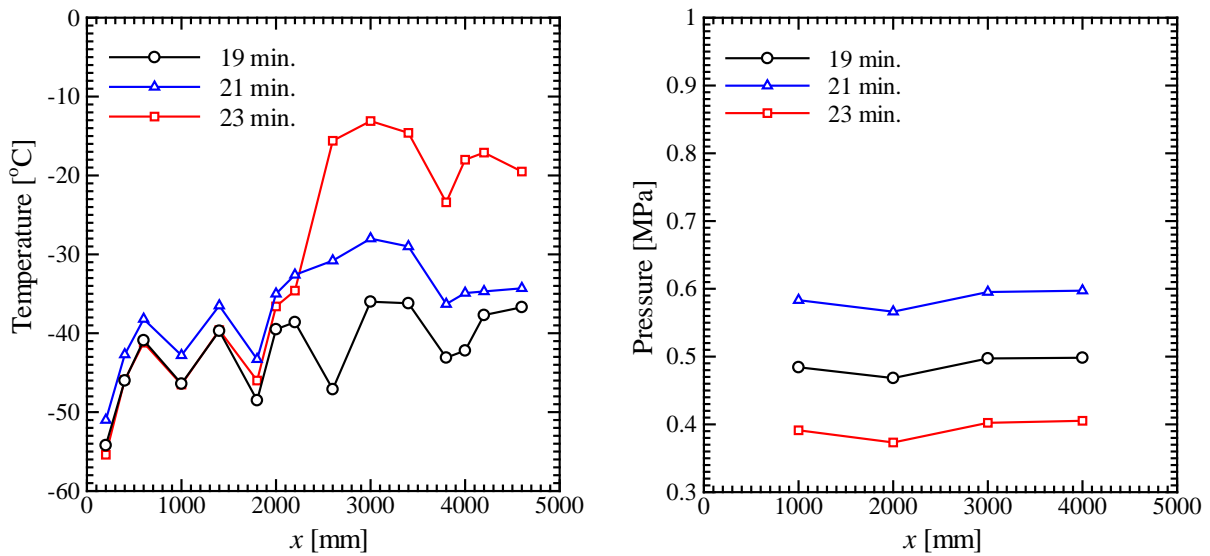
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Figure 7. Variations of measured local temperature and local pressure at different durations for the case of without swirl promoter

Figure 8 shows the variation of measured temperature and pressure values at 40, 80, and 120 min in the case of using the swirl promoter. As seen from the figure, measurement results of various local temperatures and pressures in the evaporator/sublimator at whole operation time (140 min) show almost the same value. As mentioned above for Figures 4 and 5, with the installation of the swirl promoter to the evaporator/sublimator, the system can be operated continually and stably without blockage of dry ice in the evaporator/sublimator. As clearly shown in Figure 8, the temperature increases from 0 to 2800 mm of the evaporator/sublimator with oscillation, and at 2800 mm, it starts to decrease locally, reaching -35 ℃ at 3000 mm. It is interesting to note that the temperature remains almost constant at the value of -35 ℃ until the local point of 4200 mm, and at the end of the evaporator/sublimator, the temperature slightly increases. Here, it is thought that the same amount of dry ice could be induced to the evaporator/sublimator as observed from the pressure variations remaining almost constant and lower than the CO_2

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sublimation pressure of 0.518 MPa. The increasing trend of temperatures in the range of 0 – 2800 mm is a well-known effect of developing the thermal boundary layer, indicating that the temperature difference between the inner wall and the CO_2 becomes more significant along the evaporator/sublimator. The reason for the large oscillation in temperatures between the ranges 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 [30]. After x=3000 mm, the constant temperature is maintained by solid-gas two-phase flow distribution throughout the pipe. After the sublimation process, the temperature at x=4600 mm increases due to the transformation of the dry ice into the single gaseous phase.
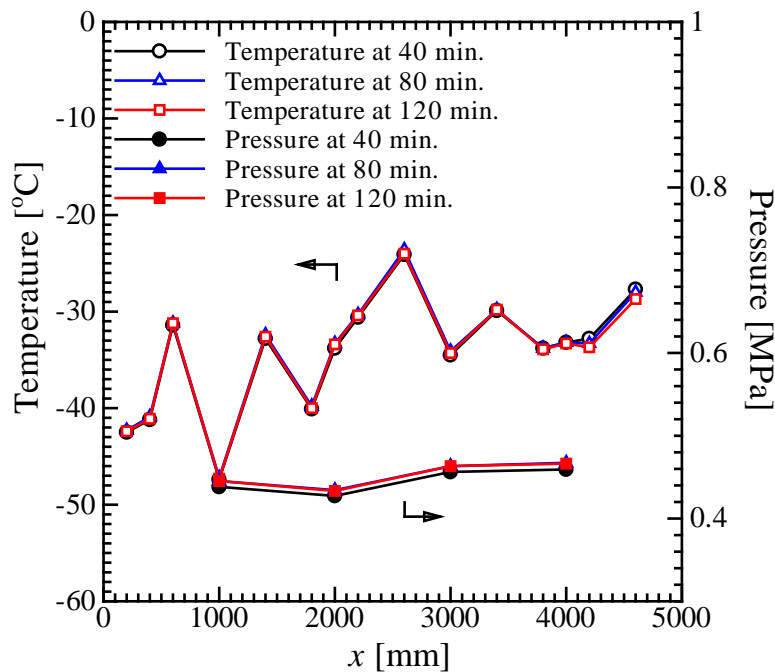

Figure 8. Variations of measured temperature and pressure values at different durations for the case of the swirl promoter.

For the discussion of the local 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 are investigated in detail, as shown in Figure 9. In the figure, the x-axis represents the position x from the inlet of the evaporator/sublimator, and the y-axis shows the local heat transfer coefficient h_x determined by the following equation;

$$h_x = \frac{q}{T_w - T_{in}} \tag{1}$$

In this equation, q is the input heat flux, T_w and T_{in} are the temperatures of the inner wall of the pipe and CO_2 passing through the evaporator/sublimator, respectively. The CO_2 temperature T_{in} was determined using the measured pressure values at the same position by PROPATH [31] and REFPROP 8.0 [32] programs. The combined standard uncertainty of the calculated heat transfer coefficient was 20 W/m^2K. As shown in Figure 9, the local heat transfer coefficient at x=1000 mm with swirl promoter is higher than that of the without swirl promoter. This is caused by inducing a swirling flow where a large number of dry ice particles may be dispersing along the inner wall of the pipe by absorbing a great deal of heat quality. In the range of x=1000 – 2000 mm, the local heat transfer coefficient for both experiments tend to decrease due to the development of the thermal boundary layer of the gaseous phase along the wall. For the swirl promoter, it is realized that the local heat transfer coefficient is increased owing to an active sublimation heat transport of dry ice. In the absence of the swirl promoter, the local heat transfer coefficient shows a decrement tendency at x=3000 mm and then starts to increase at x=4000 mm. These trends may be due to the large agglomeration of dry ice being formed in the evaporator/sublimator, where the heat transfer is degraded as the significant heat is transported via heat conduction.
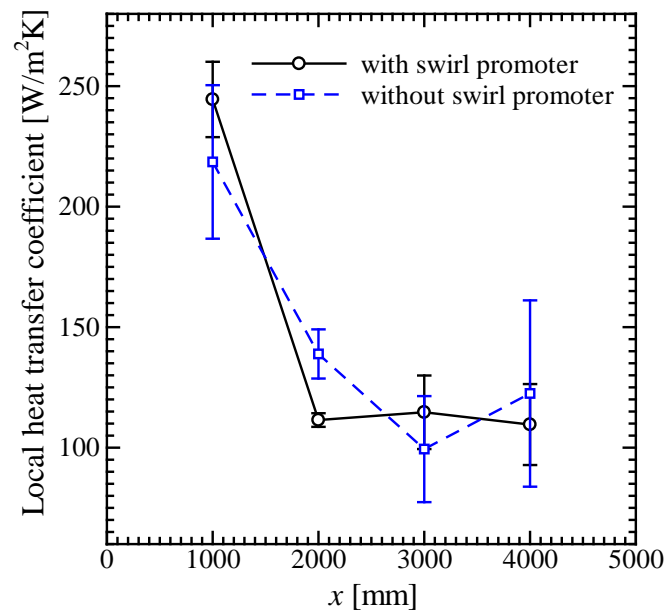
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Figure 9. Variation of local heat transfer coefficient along the evaporator/sublimator

Additionally, when the sedimentation of dry ice that fills the bottom wall of the pipe occurs, it is considered that at this condition, a 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 Figure 9 with the swirl promoter, the solid-gas two-phase transforms into the single gaseous phase, resulting in a decrement of the local heat transfer coefficient along with the evaporator, owing to developing of thick thermal boundary. On the other hand, in the absence of the swirl promoter, the local heat transfer coefficient increases at x=4000. This is due to the rise in the heat transfer coefficient because the residual dry ice sedimentation flows from upwind. Since the CO_2 heat content in the evaporator / sublimator of the cascade refrigeration system much depends on the condensation temperature at the third condenser and input heat flux, no significant difference was obtained in the amount of heat transfer characteristics as shown in Figure

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9. However, the improvement of the local heat transfer coefficient was confirmed in the second half of the evaporator tube by installing the swirl promoter.

5. Conclusions

In this study, the effect of a newly designed and constructed tapered evaporator/sublimator with the swirl promoter to the performance of the CO_2 cascade refrigeration system was investigated for ultra-low temperature applications. In order to enhance the heat transfer characters of CO_2 solid-gas two-phase flow, the experiments were carried out to observe the CO_2 solid (dry ice)-gas phenomenon in the evaporator/sublimator. In summary, the followings are concluded.

1). In the case of using the tapered channel at the entrance of the evaporator/sublimator, the blockage phenomenon was observed at the early stage of the operation time since strong dry ice agglomeration was formed at the bottom wall of the evaporator/sublimator.

2). By the installation of the swirl promoter in the evaporator/sublimator, it was observed that the system could be operated stably and continuously owing to the well developed thermal boundary layer with dry ice sublimation.

3). Dry ice particles were uniformly distributed along the inner wall of the evaporator/sublimator by the installation of the swirl promoter, where the heat transfer coefficient was largely improved.

4). Further work is certainly required to obtain more information about the effect of newly designed evaporator/sublimator for different system parameters and configurations. Also, an optimization study for future research is necessary for determining the optimum working conditions.

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Acknowledgments

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