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NEW THRUST BEARING GROOVE TO CONTROL THE OVERTURNING FOR CO_2 SCROLL COMPRESSOR WITHOUT ACCUMULATORS

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ABSTRACT

Recently, the natural refrigerants, particularly carbon dioxide (hereinafter called CO_2) with its advantageous and non-flammable properties, have been attracting attention, and now the water heaters that adopt the CO_2 heat pump systems are mass-produced in Japanese markets. This study presents a high-efficiency accumulator-less type scroll compressor with a high-pressure vessel, for CO_2 heat pump water heater system, which has a new thrust load control mechanism realizing the higher efficiency and keeping a high reliability of the thrust bearing. A new groove was made on the thrust bearing surface of the fixed scroll, in order to control the overturning of the orbiting scroll, thus preventing the gas leakage through the tip clearance, even in the low-compression-ratio operations, and thus achieving a compliance for the large amount of liquid refrigerant flows into the suction port. As a result, the accumulator-less type scroll compressor was realizable, and the compressor performance was significantly improved.

1. INTRODUCTION

The production of fluorocarbons containing chlorine is to be phased out by the year 2020 to protect the ozone layer. The Kyoto Protocol was adopted to restrict the use of HFC refrigerants from the viewpoint of high global warming potential. Thereupon, movements have occurred to review the natural substances as refrigerants for heat pumps, and recently research and development of heat pump systems, using natural refrigerants is being actively promoted. In the circumstances, particularly carbon dioxide (CO_2), with its advantageous non-flammable and non-toxic properties, is becoming a focus of attention from the viewpoint of low global warming potential, and aggressive works are being made to apply CO_2 to the heat pump water heater systems, which have already been commercialized in Japanese markets.

In this study, a high-efficiency accumulator-less type scroll compressor with a high-pressure vessel, for our CO_2 heat pump water heater system, is presented. We achieved the higher efficiency by adopting a new device of thrust load control mechanism, in which a new groove was made on the thrust bearing surface of the fixed scroll. The operating condition of the CO_2 refrigeration cycle has the feature of the low compression ratio, compared with other refrigerants. When the compressor is operated at such a low compression ratio, the orbiting scroll loses touch with the fixed scroll, so-called "overturning", and thereby the volumetric efficiency decreases caused by the increase in refrigerant gas leakage through the tip clearances. Thereupon, a special attention was paid to control the overturning of the orbiting scroll, where a new groove was made on the thrust bearing surface of the fixed scroll to increase the thrust force preventing the overturning. In order to ensure the effect of the new groove upon preventing the overturning, the relationship between the overturning limit and the suction dryness was carefully examined, experimentally and theoretically. In addition, the thrust load control mechanism realizes the accumulator-less refrigeration cycle, where the new groove and liquid release valves make a significant role to achieve compliance when the large amount of liquid refrigerant flows into the suction port, and to keep a high reliability of the sliding parts at the thrust bearing. Therefore we were able to get a success to develop the first "accumulator-less" CO_2 heat pump water heater system in the world.

2. BASIC STRUCTURE OF CO_2 SCROLL COMPRESSOR

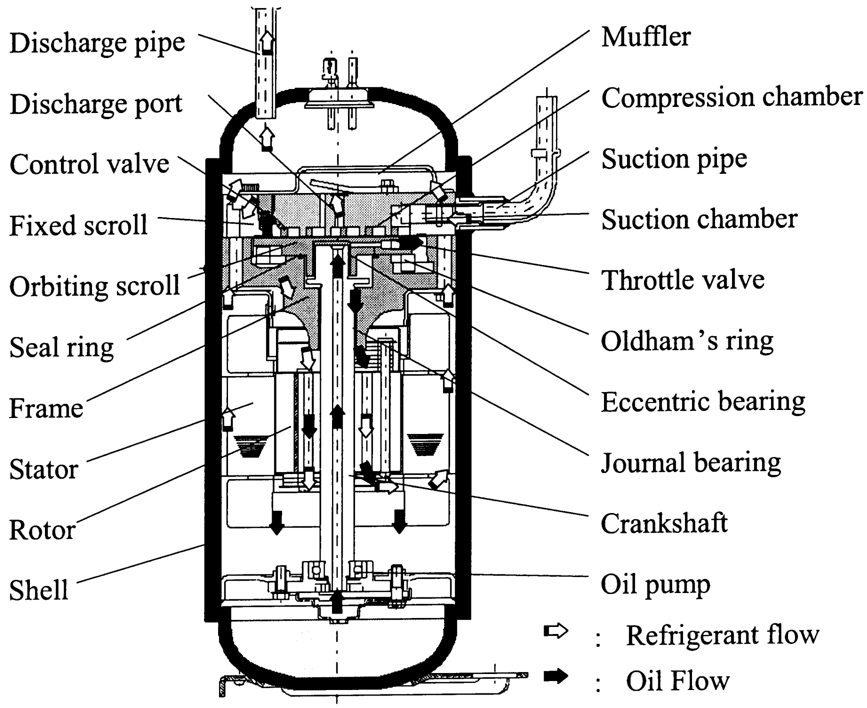

Figure 1: Cross section of the CO_2 scroll compressor

2.1 Compressor Structure

A CO_2 scroll compressor, developed by refining our scroll compressor for R410A, is shown in Figure 1. The fixed and orbiting scrolls are mated to form a multiple number of compression chambers. With the orbiting motion, these compression chambers move towards the center while reducing their volume to compress the refrigerant. The orbiting scroll is connected to the crankshaft driven by the motor. The Oldham's ring plays the role of preventing the orbiting scroll from self-rotation. The seal ring is attached between the orbiting scroll and the frame to divide the high-pressure section inside the seal ring and the intermediate pressure section outside the seal ring.

2.2 Refrigerant Flow

The refrigerant returning from the refrigerant cycle passes through the suction pipe and is introduced into the suction chambers. The refrigerant confined in the suction chambers is compressed towards the center by the orbiting motion and discharged from the discharge port. The discharged refrigerant is introduced to the lower part by the muffler and led to the upper section via the connecting channel, and discharged from the discharge pipe and sent to the refrigerant cycle. During this process, the oil in the refrigerant is separated to prevent from being discharged into the refrigerating cycle.

2.3 Oil Flow and Back-pressure Adjusting Mechanism

The oil accumulating at the bottom of the shell is pumped up by an oil pump to pass through the inside of the shaft. The oil reaching the top of the crankshaft lubricates the eccentric bearings and the main bearings in order, and then returns to the bottom of the shell.

In addition, some amount of the oil reaching the top of the crankshaft is depressurized by passing through the throttle valve installed in the orbiting scroll, and then reaches the periphery of the orbiting scroll. This oil lubricates the Oldham's ring and increases the back-pressure to prevent the orbiting scroll from overturning.

The back-pressure is maintained at the intermediate pressure by the back-pressure adjusting mechanism installed in the fixed scroll. Then the oil flows into the suction chambers, and acts to seal the gas leakages in compression process. The shortage of the oil into the compression chamber decreases the effect of the oil sealing between the compression chambers, thus resulting in a decrease of the performance. On the other hand, the large amount of the oil fed into the compression chamber induces an overheating of the refrigerant in the suction process, thus resulting in a decrease of the refrigerant circulating rate. Thereupon, our compressor made a proper setting of the throttle valve and the back-pressure to adjust the oil into the suction and compression chambers, thus achieving the higher efficiency.

3. COMPRESSOR BEHAVIORS UNDER ACCUMULATOR-LESS CYCLE

3.1 Features of "Accumulator-less" Cycle

When the accumulator is not equipped in the refrigeration cycle, not only the heat pump unit can be downsized but also the low noise can be achieved, because the noise (or refrigerant noise) transmitted from the compressor is never radiated from the accumulator. In addition, since the pressure loss in the refrigeration cycle can be reduced, the efficiency of the whole refrigerant cycle can be increased.

On the contrast, during the transient operation at starting-up, the large amount of liquid refrigerant unsteadily flows into the compressor. In addition, under the conditions in which the heat exchange capacity is reduced because of a small temperature difference between the ambient air and the evaporator, the liquid refrigerant steadily flows into the compressor. Therefore, in order to achieve the "accumulator-less" refrigeration cycle, it is necessary to increase the compressor reliability and the dynamic stability for both unsteady liquid return and steady liquid return.

3.2 Case of Unsteady Return of the Liquid Refrigerant

Figure 2 and 3 show the time history of the discharge and suction pressures at starting-up at a low ambient temperature with and without an accumulator. Focusing on the discharge pressure in these figures, it was found that when an accumulator is equipped, the discharge pressure rises smoothly, while in the case of accumulator-less, the discharge pressure suddenly rises when the large amount of liquid refrigerant flows into the compressor. Furthermore, it is confirmed that the wear of the bottom surface of the orbiting scroll increases because the orbiting scroll is strongly pressed against the fixed scroll by the sudden increase of compression ratio.

To solve this problem, the release valves are installed to discharge the liquid refrigerant. As shown in Figure 4, the release valves are made on the place where the liquid refrigerant can be released immediately after the chambers finish suction process. With this configuration, when the pressure abnormally rises by liquid compression, the refrigerant can be quickly released.

Figure 5 shows the time history of the discharge and suction pressures at starting-up at a low ambient temperature without an accumulator when the release valves shown in Figure 4 are installed. Focusing on the discharge pressure in this figure, it is confirmed that the discharge pressure smoothly rises, as well as the case in which an accumulator is installed. In addition, the wear of the bottom surface of the orbiting scroll was not confirmed, and the sliding part was under a better condition.

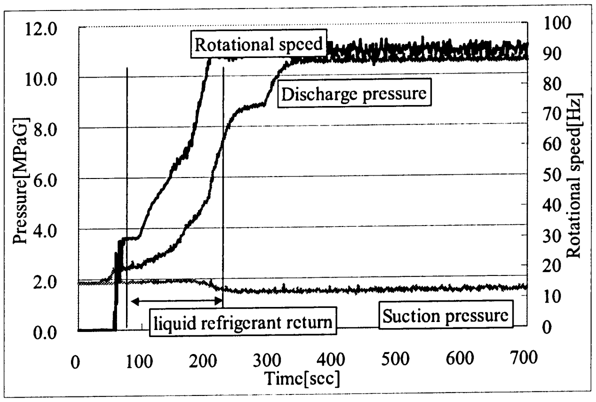

Figure 2: Trend graph of discharge and suction pressures with an accumulator

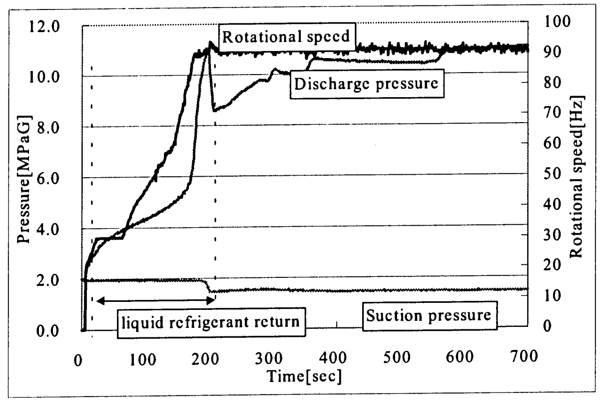

Figure 3: Trend graph of discharge and suction pressures without accumulators

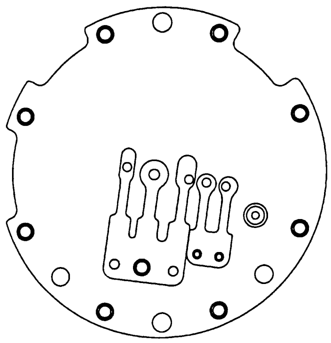

Figure 4: Liquid release valve

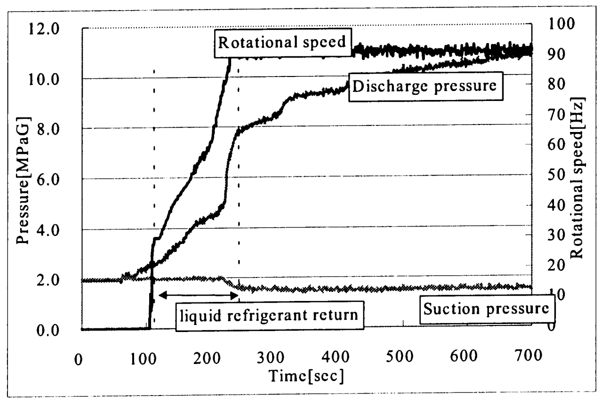

Figure 5: Trend graph of discharge and suction pressures with the liquid release valve without accumulators

3.3 Case of Steady Return of the Liquid Refrigerant

On the other hand, under the conditions in which the heat exchange capacity is reduced because of a small temperature difference between the ambient air and the evaporator, the liquid refrigerant steadily flows into the compressor. Therefore, when the scroll compressor with the thrust load control mechanism is adopted, a careful attention should be paid to low compression ratio conditions, including "overturning", which disengages the orbiting scroll from the fixed scroll and decreasing the volumetric efficiency, as described above. Moreover, in the case when the liquid refrigerant steadily returns into the compressor, the "overturning" is confirmed at a comparatively high compression ratio.

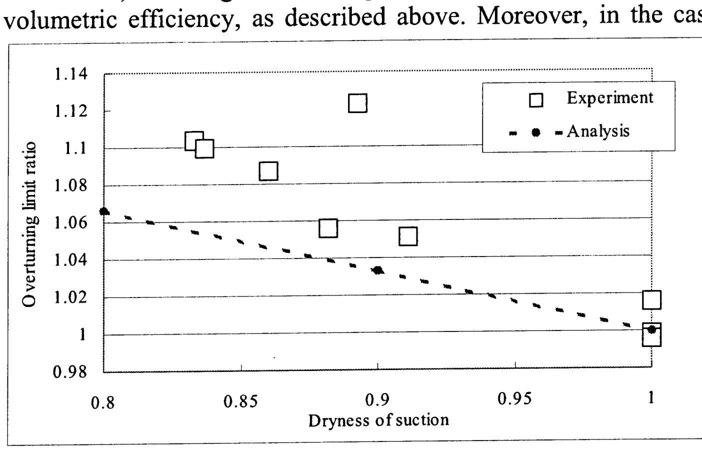

Figure 6: Relationship between the critical overturning limit ratio and dryness of the suction

Figure 6 shows the experimental and analytical results of the relationship between the overturning limit of compression ratio including the "overturning" and the dryness of the suction. This figure clearly indicates that as the dryness decreases, the overturning limit of compression ratio increases for both the experimental and analysis results. In addition, it also indicates that when the dryness approaches in the vicinity of 0.8, the overturning critical compression ratio increases by about 10% than the case in which the dryness is 1.0. In other word, the compressor overturns easily as the dryness of the suction decreases. This phenomenon is explained as follows: as the dryness of the suction decreases, the gradient of isentropic line on the P-H diagram increases, and hence the pressure rise in the compression process is accelerated, thus increasing the force to disengage the orbiting scroll from the fixed scroll.

Based on these results, investigations can be made on how to control the overturning when the liquid refrigerant steadily flows into the compressor.

4. CONTROL OF OVERTURNING

4.1 Oil Supply Delay

Our scroll compressor is equipped with the back-pressure adjustment mechanism as described above. As shown in Figure 7, in order to suppress "overturning" disengaging the orbiting scroll from the fixed scroll because of the small thrust force, one may increase the back-pressure. However, when the back-pressure increases, the oil is not fed from the back-pressure adjusting mechanism into the compression chambers while the discharge pressure rises to the back-pressure. As a result, an oil supply delay in lubrication to the compression chambers occurs.

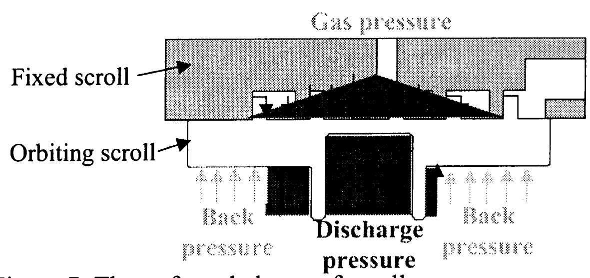

Figure 7: Thrust force balance of scroll compressor

Figure 8 shows the relationship between the back-pressure and the time of oil supply delay at starting-up. As shown in this figure, when the back-pressure increases to 0.5, 1.0 and 1.5 MPa, the time duration when the oil is not fed into the compression chambers increases.

The time duration of non-lubricating to the compressor exerts a serious effect on the compressor reliability. Therefore, in the heat pump water heater systems, the start-up tests are carried out at the low ambient temperature when the large amount of liquid refrigerant flows into the suction port of the compressor. Comparing the wear at 1.5 MPa back-pressure with at 1.0MPa back-pressure, the scratched damages were more severe, confirmed in the part of the bottom surface of the orbiting scroll when the back-pressure was set to 1.5 MPa, as shown in Figure 9. However, when the back-pressure was set to 1.0 MPa, there appeared no scratched damages on the bottom surface of the orbiting scroll. In addition, any wear of the bottom surface of the orbiting scroll was not confirmed, and the sliding condition was under very well.

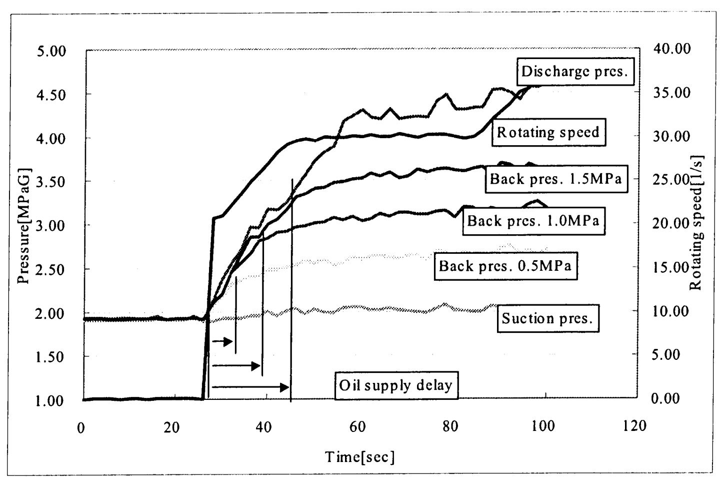

Figure 8: Trend graph of the back-pressure at the time of startup

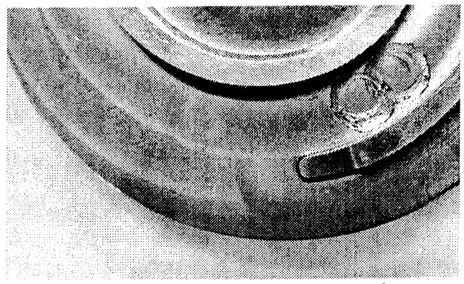

Figure 9: Scratched damages on the orbiting scroll

Based on these results, the thrust load control mechanism that were able to control the overturning were modified, securing the reliability by setting the back-pressure to 1.0 MPa or lower.

4.2 New Groove on the Thrust Bearing

One may devise another method that makes possible to suppress the "overturning" disengaging the orbiting scroll from the fixed scroll, where the orbiting scroll is pulled toward the fixed scroll. Figure 10 shows the thrust bearings on which a groove is formed to pull the orbiting scroll. With respect to Specification A, a groove that occupies about 37% of the total thrust bearing area is formed, whereas for Specification B, a groove that occupies about 15% is formed. In addition, since these grooves are formed inwards from the orbiting trajectory of the edge of the orbiting scroll, these are isolated from the back-pressure chamber. Furthermore, since these are also isolated from the compression chambers in the compression process, the grooves on the thrust bearing of the shaded area of Figure 10 are held at the suction pressure.

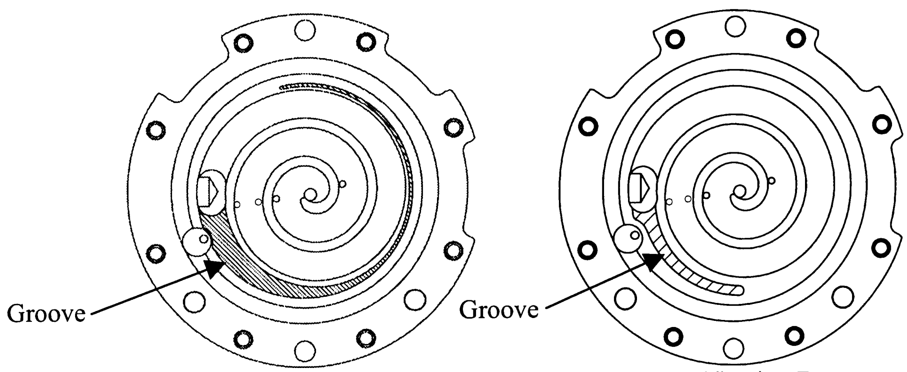

Figure 10: Grooves on the thrust bearing to control the overturning

4.2 Experimental Results

The overturning limit of the compression ratio was measured for no groove and grooves of Specification A and B, as shown in Figure 11, where the abscissa is the back-pressure. From the experimental results for no groove, the overturning limit for Specification A and Specification B was also calculated, as shown in the same figure, where the vertical force balance acting on the orbiting scroll was taken into consideration, assuming that the oil film pressure of the thrust surface was same as the back-pressure and the pressure of the groove on the thrust bearing was same as the suction pressure. The dryness of the suction was assumed to be 1.0.

As clearly shown in Figure 11, the groove on the thrust bearing greatly improves the overturning limit. Now, assume the target of the overturning limit of compression ratio to be 1.3. In order to achieve this compression ratio of 1.3, the back-pressure of the groove of Specification A is 0.8 MPa, whereas 1.2 MPa for the groove of Specification B and 1.35 MPa for no groove. To shorten the time of the oil supply delay, the groove of Specification A is most effective. When the groove of Specification A is adopted, the heat pump water heater system can be operated at compression ratio of 1.3 or higher, controlling the overturning. If the overturning limit of the compression ratio decreases by about 10% caused by the effects of the steady liquid return, it is possible to operate at compression ratio of 1.4 or higher. If the compression ratio is getting 1.4 or higher, almost all the operating conditions of the heat pump water heater system can be covered.

As shown Figure 11, the overturning limit of the experimental results for Specification A and Specification B is improved than that of the calculated results. It is assumed that the groove suction pressure has the influence on the oil film formation on the thrust surface. Thereupon, the suction pressure area on the thrust bearing was equivalently increased from the area of the groove.

The compressor efficiency of Specification A is compared with no groove, as shown in Figure 12. There appears no effect of the groove on the thrust bearing for the compressor efficiency under the rated conditions (compression ratio: 2.1) of the heat pump water heater system.

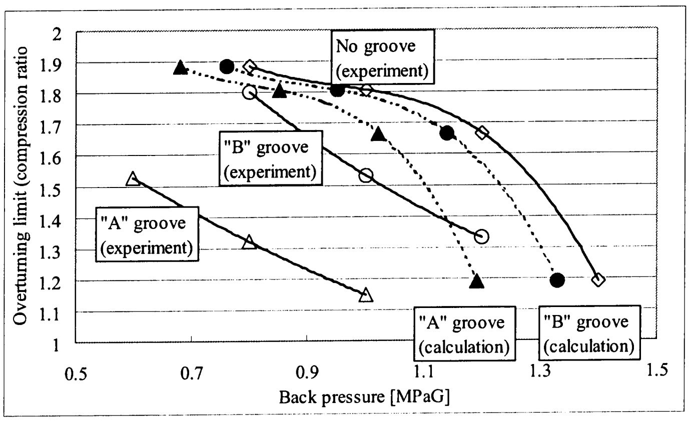

Figure 11: Relationship between back-pressure and overturning limit

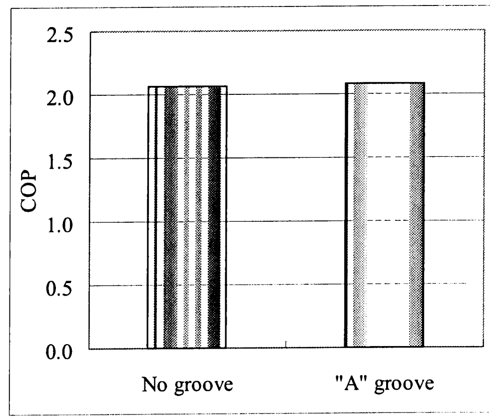

Figure 12: Groove influence on the efficiency

6. CONCLUSIONS

The scroll compressor for CO_2 heat pump system without accumulators was developed and the following conclusions were obtained.

- A high reliability was secured for unsteady and steady liquid returns even if without accumulators.
- The groove on the thrust bearing made possible to operate the scroll compressor at a high efficiency also in low compression ratio.

As a result, the noise reduction and space saving have been achieved, both of which are significant features of "accumulator-less" refrigeration cycle. Furthermore, the highest COP at present has been achieved in CO_2 heat pump water heater system by the synergy effect of the small pressure loss due to the accumulator-less refrigeration cycle and the adaptability for CO_2 heat pump systems due to the thrust load control mechanism.

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