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EVAPORATION OF LIQUID PARTICLES IN TWO-PHASE COMPRESSION PROCESS

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ABSTRACT

The change in water particle diameters injected into the vapor compression process, the water phase temperature and the vapor phase temperature were measured in the compression process. The experimental conditions were approx. 2 at a compression ratio under 100 kPa in pressure, 340~390 K in temperature range, 2~8 Hz in rotational speed. The mass ratio of injected water was small: in the order of 1% of the vapor phase mass.

The temperature of water particles was clarified as being the same as the saturation temperature for the measured pressures. The temperature difference between two phases was in the order of 10 K under the test conditions. Water particles with diameters of some ten micrometers were sufficiently evaporated in the compression cycle. Injection timing difference was clearly apparent in the temperature-entropy diagram at the rotational speed of 2 Hz, even for very small injection mass ratio.

INTRODUCTION

High temperature process heat is usually supplied by boilers. If heat pump systems that can supply high temperature heat output are developed and become widely used, this would impact substantial effect on energy conservation.

Although, because of its thermal stability, water is suitable as the working fluid for high temperature compression heat pumps, adiabatic compression of steam from a saturated condition induces a large degree of discharge superheat. This not only reduces the coefficient of performance (COP), but in some cases, heat pump operation becomes impossible owing to the temperature limit of some materials.

The two-phase compression process accompanied by liquid refrigerant injection cooling is the key technology for realizing high performance high temperature heat pump systems. The authors have theoretically shown COP improvement in the case where the refrigerant is water on cycle analysis assuming the two-phase compression process to be quasi-static1).

Two-phase compression processes in real compressors are not quasi-static due to the relaxation process of evaporation of liquid phase. The liquid phase injected into the vapor phase is atomized, and the liquid particles and vapor phase deviate from their equilibrium temperature. This deviation introduces irreversibility in the compression process and results in COP reduction compared with the ideal process where the liquid phase and the vapor phase are in thermal equilibrium. Although, the sizes and the evaporation speed of the liquid particles are important parameters in utilizing the two-phase compression process successfully, there are no studies measuring the effects of these parameters on liquid refrigerant injection cooling systems.

In this paper, we measured the change in water particle diameters injected into the vapor compression process, and at the same time, we attempted to measure the temperature difference between the water phase and the vapor phase.

EXPERIMENTAL APPARATUS

The experimental apparatus is shown in Fig. 1.

Chambers and Injectors

The pressure chamber was composed of two parts, the test section and compression cylinder. Both parts were connected to each other by a pipe approx. 10 cm long and 6 mm in inner diameter. Total inner volume of both parts changed from 522 cm^3 at the bottom dead center to 233 cm^3 at the top dead center of the piston. These values were estimated from the static pressure rise of dry air at room temperature between both dead centers and the stroke volume, all of which could be measured accurately.

The temperature of the test section walls is automatically controlled to approximately 90 ^{o}C (363 K) by winding electric tape heaters around the walls. The cylinder has a hot water jacket and its temperature is also automatically controlled to approximately 92 ^{o}C (365 K).

The pressure in the test section changes periodically owing to the reciprocating motion of the piston in the compression cylinder. Repeating cycles of compression and expansion occur with constant mass of vapor in the chamber if there is no liquid injection or exhaust.

Two injectors of the type generally used for electronic fuel injection systems of vehicle gasoline engines, were attached to the test section. One was on the top plate and the other at the bottom plate of the test section. A small amount of the liquid was injected into the test section by the upper injector. The lower injector was used as an exhaust valve. The outside of the lower injector was under vacuum. The average pressure in the chambers was determined by the balance of the injection mass flow rate and the exhaust mass flow rate during a cycle.

The compression strokes of the test section simulated the compression stroke of real compressors. Only the quantities of the vapor state in the test section were measured and discussed, and there was no need to consider those in the compression cylinder. That is, the vapor flow in the connecting pipe was considered as a vapor piston.

The piston in the compression cylinder has two stages and a compartment between these stages. This compartment is kept under vacuum to avoid air leakage into the chamber through the piston rings.

Test Fluid

The test fluid used was ion-exchanged water.

Particles Measurement System

To measure the diameters and velocities of the water particles, a Doppler Signal Analyzer (DSA)[2)] of Aerometrics, Inc., was used. The point of particle measurement was settled at the center of the test section. The evaporation speed of the particles was obtained from the averaged time dependence of the diameters of the measured particles.

Thermometers

Two types of thermometers were used in trying to measure the temperatures of the liquid phase and the gas phase: A K-thermocouple 15 μm in diameter and a resistance thermometer using a cold wire probe 2.5 μm in diameter and 5 mm in length.

The flow of the vapor from the connecting pipe into the test section was superheated slightly more than the vapor in the test section, owing to the friction in the pipe flow. The resistance thermometer was fixed in the area where this flow mixed into the vapor in the test section. The resistance thermometer was expected to show the temperature of the vapor phase because the surroundings contain fewer water particles. The thermocouple was settled at a point where additional water particles would strike it, and thus kept it in wet condition longer. The thermocouple was assumed to show the temperature of the water phase when it was wet.

EXPERIMENTAL RESULTS

The saturation pressure of the water was approx. 70 kPa at 90 °C (363 K), with the wall temperature, and the temperature and pressure conditions of the tests were near those values.

The two thermometer temperatures (**Tk**: thermocouple, **Tr**: resistance thermometer), pressure, and diameters and velocities of the water particles were measured with crank angle rotational speeds of 2, 4, 8 Hz. The crank angle synchronizes the data obtained in continuous cycles as the data of one cycle. The values of 0 and 180 deg in crank angle show the bottom dead center (BDC) and the top dead center (TDC), respectively.

Since a smaller amount of injection mass flow rate was more convenient for clearly measuring the evaporation of the particles, the experiments were done with a short injection time period. An electric signal period of 4 msec was fixed as injection time. The injection mass flow rate changed with injection pressure difference and the electric signal period for injection as shown in Fig. 2. Mass flow rate decreased with increasing injection pressure difference in such a short electric signal period, because the injection time delay increased. Under experimental pressure conditions, the time delay was approx. 3.5 msec, with the real injection period approx. 0.5 msec. The injection mass flow rate is small: in the order of 1% of the mass of the vapor phase.

The experimental results concerning the water particles are shown in Fig. 3 to Fig. 11. Each figure represents a particle count of approx. 2000.

The particles were classified into two groups, the particles measured immediately after injection and the particles that were moving with the vapor. Some portion in the latter group may have yielded in the expansion strokes, with some portion coming from the former group. The particles in the latter group probably existed almost equally in the test section, and the lack of appearance of the particles in the high pressure area shows the complete evaporation of the injected particles.

The experimental results concerning pressure and temperature for the rotational speed of 4 Hz are shown in Fig. 12. **Ts** is the saturation temperature for the measured pressure. **Tad** is the calculated temperature with the adiabatic pressure change being the same as the measured pressure change.

The temperature of the water particles was clarified to be the same as the saturation temperature for the measured pressures, because the thermocouple showed a very close value to the saturation temperature for the measured pressures when it was considered to be wet. Further, **Tr** changed in the same manner as **Tad** and it showed the vapor temperature. The temperature difference between the liquid phase and the vapor phase was in the order of 10 K.

Figure 13 is the temperature-specific entropy (**T-s**) diagram of the vapor phase in the compression processes with a different injection timing of 1.6 and 90 deg crank angle, at a rotational speed of 2 Hz. The different injection timing clearly appears on the **T-s** diagram even for very small injection mass ratio. This shows that well-controlled injection timing will realize the optimum timing conditions expected by theoretical analysis1).

CONCLUSIONS

The change in the water particle diameters injected into the vapor compression process, the water phase temperature and the vapor phase temperature were measured in the compression process. The experimental conditions were approx. 2 at a compression ratio under 100 kPa of pressure, 340~390 K in temperature range, and 2~8 Hz in rotational speed. The mass ratio of injected water was small: in the order of 1% of the vapor phase mass.

As results, the temperature of the water particles was clarified as being the same as the saturation temperature for measured pressures, because the thermocouple showed a very close value to the saturation temperature for measured pressures when it was assumed to be wet. The temperature difference between the liquid phase and the gas phase was in the order of 10 K under the test conditions. Water particles with diameters of some ten micro meters were sufficiently evaporated out in the compression cycle. The difference in the injection timing clearly appeared on the temperature-entropy diagram at a rotational speed of 2 Hz, even with very small volumes of injection.

Although the experimental conditions were different from those in real compression processes of high temperature heat pump systems, the experiments in this paper were useful for realizing the two phase compression process and in confirming the theoretical treatment of the two phase compression process.

REFERENCES

1) Ozaki,K., Endo,N., Yabe,A. and Kobayashi,T., "Basic Study on High Performance Heat Pump Systems Accompanying Two-phase Compression Process", Proceedings of the 1990 International Compressor Conference - At Purdue, Vol.1, pp.183.

2) Bacharo,W.D.,Werthimer,D..Raffanti,R.,and Hermes,R.J.,1989,"A High Speed Doppler Signal Processor for Frequency and Phase Measurements",Third Int. Conf. on Laser Anemometry, Advances and Applications, Swansea, United Kingdom,September.

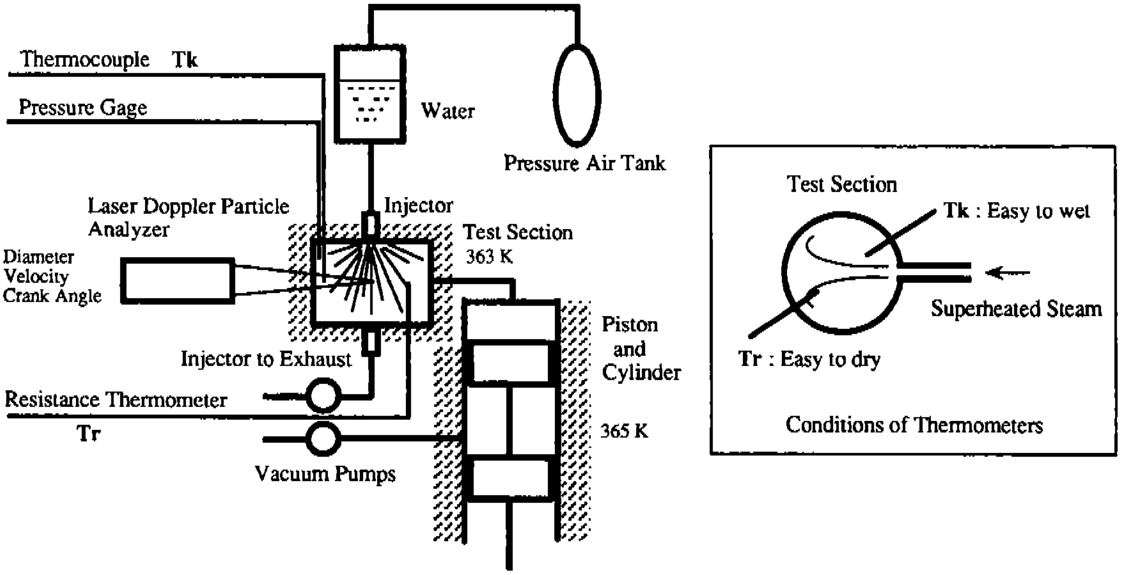

Fig.1 Schematic Figure of Test Equipment

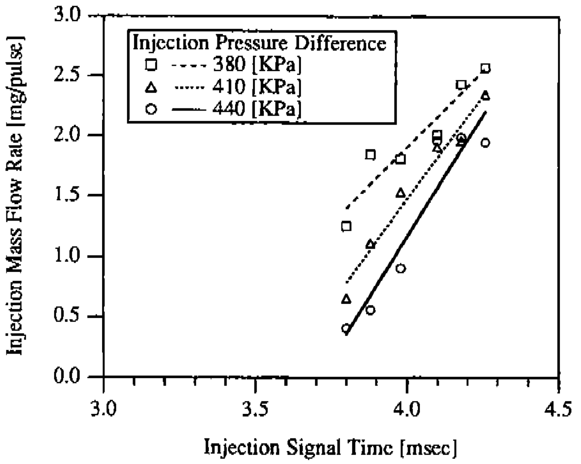

Fig. 2 Injection Mass Flow Rate and Injection Signal Time

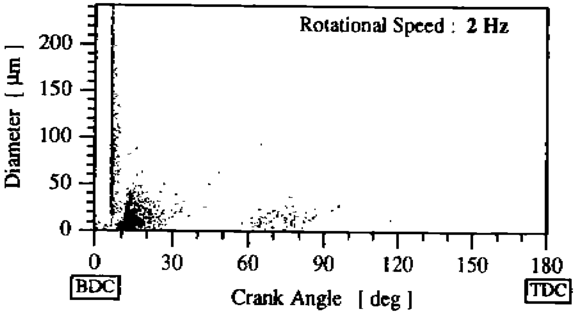

Fig. 3 Diameter Change in Compression Process

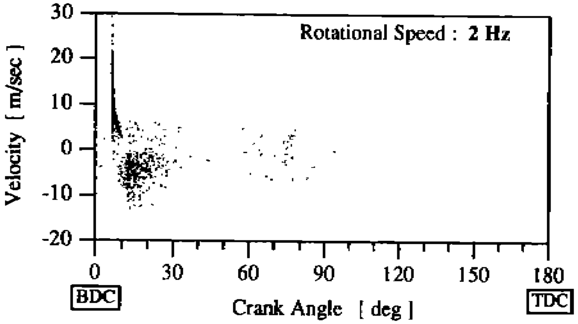

Fig. 4 Velocities of Particles in Compression Process

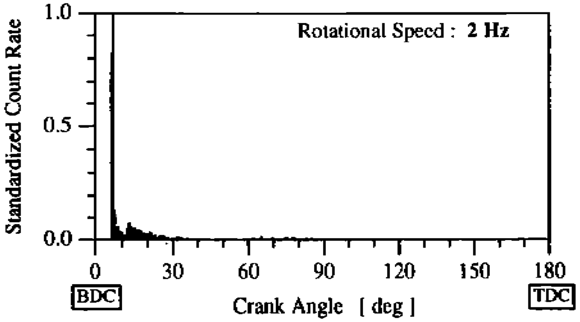

Fig. 5 Count Rate Distribution in
0.5deg Crank Angle Increment

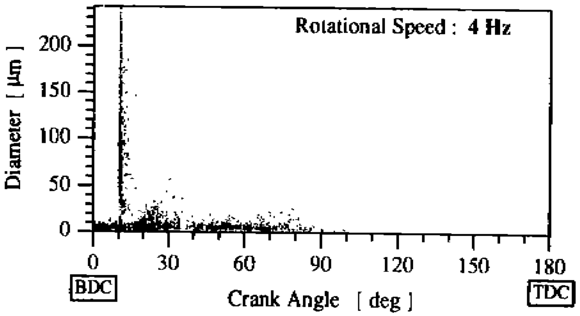

Fig. 6 Diameter Change in Compression Process

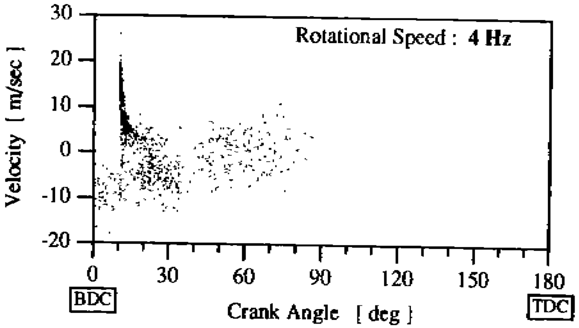

Fig. 7 Velocities of Particles in Compression Process

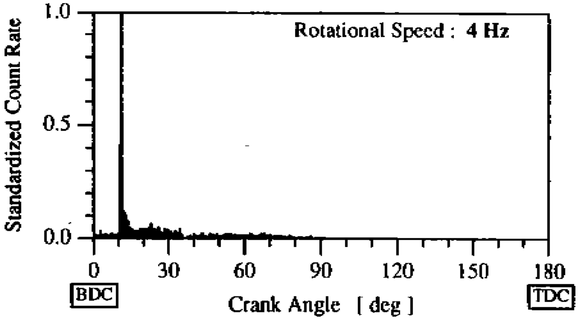

Fig. 8 Count Rate Distribution in
0.5deg Crank Angle Increment

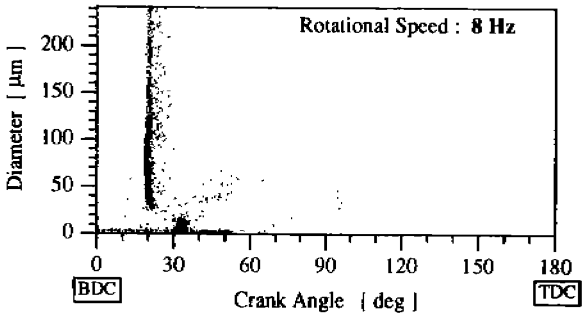

Fig. 9 Diameter Change in Compression Process

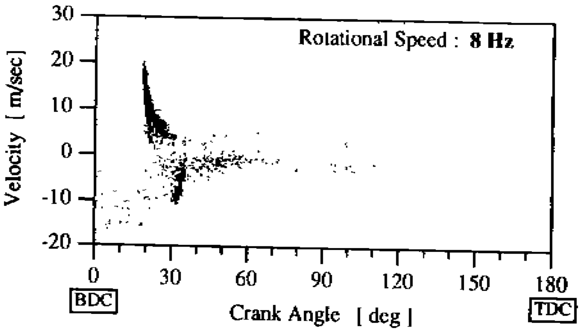

Fig. 10 Velocities of Particles in Compression Process

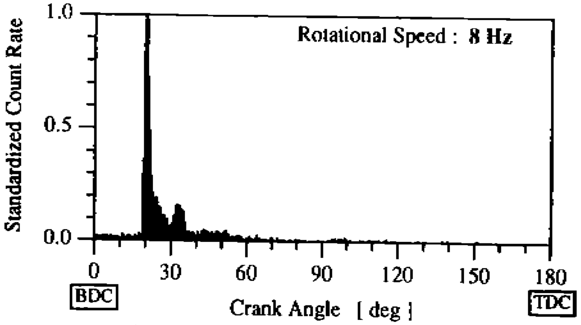

Fig. 11 Count Rate Distribution in
0.5deg Crank Angle Increment

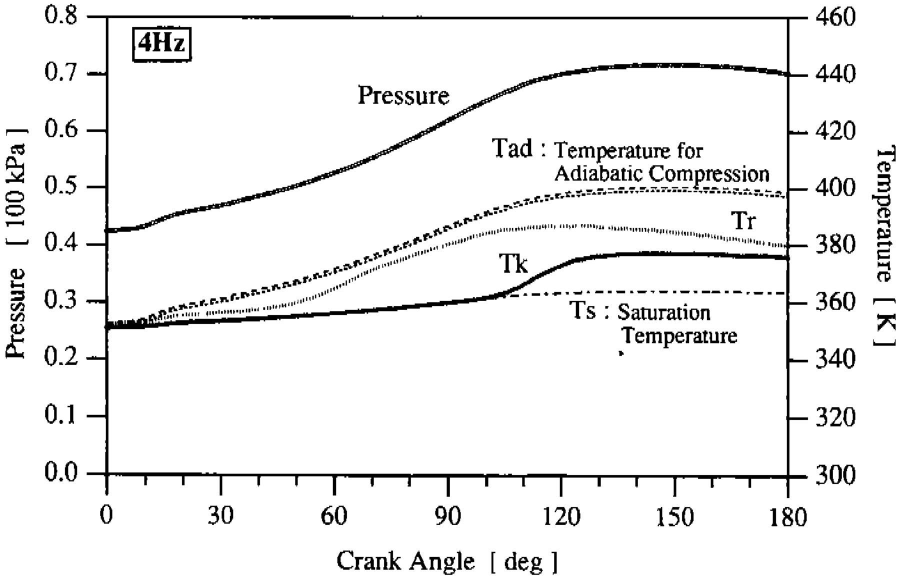

Fig. 12 Pressure and Temperature Change in Compression Process

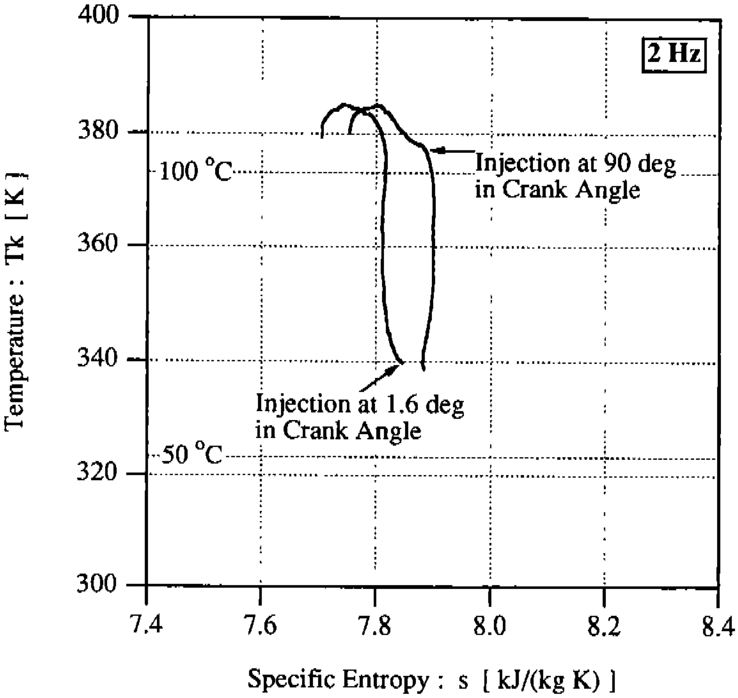

Fig. 13 Temperature-Entropy Diagram in Compression Process, Influence of The Injection Timing