Continuous Cooling Compressor for water refrigerant heat pump

T. Shoyama¹,², H. Sun¹, B. Kawano¹ and M. Matsui¹

¹Appliances Company, Panasonic Corporation, Moriguchi, Osaka, 570-8501, Japan
²shoyama.tadayoshi@jp.panasonic.com

Abstract. A turbo compressor was developed for water refrigerant heat pumps equipped with continuous cooling compression, in which the vapour is continuously cooled during the compression process by the latent heat of evaporation of the liquid water refrigerant sprayed into the impeller channels. One of the disadvantages of water refrigerant is that the discharge temperature during vapour compression is higher than that of fluorocarbon refrigerant. Multi-stage compression with intercooling are one of the solutions, the system becomes large. In continuous cooling compression, fine particles of water are sprayed from a nozzle embedded inside the impeller, and the water vapour in compression process is directly cooled to the saturation temperature. Therefore, if sufficient amount of spray is present, the compression process follows the saturation line in p-h diagram. Water is supplied to the spray nozzle after pressurized to 10 MPa or more using the centrifugal effect of rotor rotation. A small nozzle was developed that suppresses rotor imbalance and can withstand the load of centrifugal force. A followability index was defined as a reference for the droplets to follow the vapour flow and evaporate before colliding with the blade surface. A water-refrigerant heat pump system was constructed, and the degree of superheat of the discharged vapour was evaluated while changing the spray amount. Increasing the amount of spray decreased the superheat but COP also decreased. Saturated compression was demonstrated above a certain amount of spray, but the impeller flow was choked due to the droplet evaporation. It is desirable to assume a sufficient choke margin, or to develop a special CFD code that can take into account the droplet evaporation.

1. Introduction

Water refrigerant heat pump has high theoretical COP and environmental friendliness ¹. Its applications are not limited to chillers, but also include ice generation ² and high-temperature heat pumps ³. Commercial products have been released such as Efficient Energy’s eChiller and Kawasaki Heavy Industries’ MizTurbo. However, due to the high specific volume of water refrigerant, the equipment dimension tends to be large, and the cost of the compressor becomes high ⁴. Therefore, a low-cost and high-performance compressor is necessary. We have developed a water-refrigerant system with a two-stage turbo compressor that uses water as both a refrigerant and a lubricant, and achieved COP = 5.0 ⁵. One of the difficulties in a water-refrigerant turbo compressor is a vibration of the rotating system and bearings. Thus it is preferable to use water-lubricated bearings, which are simple, low cost, and have a long life ⁶,⁷.

Another difficulty is a high degree of superheat during steam compression. Fluorocarbon refrigerants, being designed for the refrigeration cycle, has an adiabatic compression line increasing along the
saturation line in a p-h diagram, reducing the theoretical compression energy. On the other hand, water has an adiabatic compression line increasing with a low gradient. Therefore, lowering the discharge temperature by intercooling by mist spraying leads to low theoretical compression energy. This has been theoretically shown and demonstrated also in our previous study. In the actual vapour compression process, droplets mixed in the suction flow of the compressor causes erosion of the impeller and a decrease in surge margin, whereas the compressed hot vapour is cooled by the latent heat of vaporization of the droplets, resulting in a compression process close to saturation, and the compression power is lowered. Wu et al. showed that an initial droplet size must be less than 10 μm for complete evaporation of the droplet and is independent on the relative spray velocity. However, for atomization to the droplet size of 10 μm or less, water must be supplied at a high-pressure exceeding 10 MPa to the nozzle. A pump capable of such high pressure is too large and too energy-consuming to be equipped with separately. In this paper, continuous cooling compression (CCC) is proposed, in which sprayed mist with an average particle size of 5.8 μm was achieved using the centrifugal effect of the impeller. The fine mists are sprayed from the impeller, with low relative velocity to the blade, suppressing the erosion. The theory and the experimental results are reported.

2. Theory of continuous cooling compression

2.1. Thermodynamics

Here CCC is defined as a thermodynamic process. Nozzles are embedded in rotating impellers. They spray water droplets that is fine enough to accompany the compressed vapour flow between the blades. The vapour is cooled during the compression process by the latent heat of vaporization of the droplets. The thermodynamic process of CCC is depicted in figure 1 as p-h and p-v diagrams.

![Diagram showing thermodynamic processes of CCC](image)

Figure 1. Thermodynamic diagrams of CCC

In the case of a two-stage compression process with intercooling (IC), compressed vapour is cooled close to the saturation temperature in the isobaric cooling process (2-3) between the first-stage adiabatic compression (1-2) and the second-stage adiabatic compression (3-4). The required cooling heat is \( c_2 = h_3 - h_2 \), and the total theoretical compression power is expressed by the following equation, ignoring the change in entropy.

\[
\Delta h_2 = (h_2 - h_1) + (h_4 - h_3) = \int_{P_1}^{P_2} v_{ad} dP + \int_{P_3}^{P_4} v_{ad} dP
\]  (1)
where \( v_{\text{ad}} \) is the specific volume in adiabatic compression, and \( \Delta h_2 \) is equal to the hatched area on the p-v diagram. It should be noted that the subscript 2 of \( c_2 \) and \( \Delta h_2 \) is the number of stages. When it is increased to \( n \), the theoretical compression energy \( \Delta h_n \) and the intercooling heat \( c_n \) are expressed by the following equations.

\[
\Delta h_n = \sum_{i=1}^{n} (h_{2i} - h_{2i-1}) = \sum_{i=1}^{n} \int_{p_{2i-1}}^{p_{2i}} v_{\text{ad}} dP \quad (2)
\]

\[
c_n = \sum_{i=1}^{n-1} (h_{2i} - h_{2i+1}) \quad (3)
\]

CCC process is the limit of \( n \to \infty \). As shown in figure 1, the process is along the saturation line. The theoretical compression energy \( \Delta h_\infty \) is the limit of Eq. (2).

\[
\Delta h_\infty = \sum_{i=1}^{\infty} \int_{p_{2i-1}}^{p_{2i}} v_{\text{ad}} dP = \int_{p_1}^{p_1} v_{\text{sat}} dP \quad (4)
\]

where \( v_{\text{sat}} \) is the specific volume in saturation, \( \gamma \) is the pressure ratio. Just like \( \Delta h_2 \neq h_2 - h_1 \) in two-stage compressor, \( \Delta h_\infty \neq h_2 P_1 - h_1 \) in CCC. Therefore, the theoretical compression power of the CCC process must be calculated from the saturation line on the P-v diagram, or the calculation of Eq. (2) must be performed for a sufficiently large number of \( n \). When the calculation was performed, it converged in 100 steps. The area between the adiabatic line and the saturation line shown in the P-v diagram is the theoretical limit of the power reduction effect by CCC, which is expressed as follows.

\[
\int_{p_1}^{p_1} v_{\text{ad}} dP - \int_{p_1}^{p_1} v_{\text{sat}} dP = \int_{p_1}^{p_1} (v_{\text{ad}} - v_{\text{sat}}) dP \quad (5)
\]

Thus, theoretical work in CCC is always lower than those in the intercooled compression. In the present case, 5% reduction of theoretical work was estimated.

### 2.2. Centrifugal pressure spraying

There are two types of method to spray droplets for CCC: external spraying and centrifugal pressure spraying. In the former, mist is generated outside the compressor and sprayed into the compressor suction path together with the vapour. This is used in wet compression technology \(^9\) and advanced humid air turbine (AHAT) systems \(^10\). This method is simple and has similar performance gain, but has two problems. The first is erosion at the leading edge of the blade due to the droplet collision because the relative velocity of the droplet to the blade is high. The second is necessity of a high-pressure pump of over 10 MPa that allow to generate sufficiently fine droplets. Centrifugal pressure spraying solves these problems at the same time by spraying from a nozzle embedded inside a rotating impeller as shown in figure 2. Water is supplied from an axial flow path through the centre of the axis connected radial flow paths in the impeller. Nozzles are attached to the tip of the radial flow path, to which the water is supplied and pressurized by centrifugal effect of the rotation. This axial flow path is shared with the centrifugal pressurization and supplying to the water-lubricated bearing. The low relative velocity between the droplet and the impeller, and the high-speed rotation of the impeller provides both of sufficiently high pressure and no erosion. Despite a problem of vibration due to increased imbalance due to the existence of flow paths and nozzles in the rotating body, the centrifugal pressure spraying was adopted in this study, by solving them by the symmetric arrangement conditions described later.
In order to avoid cavitation inside the axial flow path, it is necessary to supply water with positive pressure $P_s$ from the outside. Assuming that the rotational angular velocity of the rotor is $\omega$, the radius position of the centrifugal spray nozzle is $r_n$, the radius of the axial flow path is $r_0$, and the density of water is $\rho$, the centrifugal pressure applied to the nozzle is given by the following equation.

$$P_n = P_s + \frac{1}{2} \rho \omega^2 (r_n^2 - r_0^2) \approx P_s + \frac{1}{2} \rho (\omega r_n)^2$$  \hspace{1cm} (6)

Here, the approximation holds because when $r_0 \ll r_n$. The pressure loss in the flow path is negligible compared to $P_n$. Assuming that the spray volume flow rate is $Q_n$, the centrifugal spray power is calculated by the following equation.

$$W_n = (P_n - P_s) Q_n = \frac{1}{2} Q_n \rho (\omega r_n)^2$$ \hspace{1cm} (7)

2.3. Particle traceability

If the particle size is too large to follow the vapour flow, it will collide with blades adjacent to the nozzle or shrouds. The colliding droplets don’t effectively contribute to the continuous cooling. Here we consider the relation between particle size and its followability to the vapour flow.

Figure 3 shows the motion of droplets sprayed from a nozzle at the hub surface of a rotating impeller in a rotating coordinate system. It is assumed that the droplets are sprayed into the water vapour flowing through between the blades at a relative velocity $U$. Centrifugal force accelerates the particles in almost the same direction as the steam flow, so the effect on impeller collision is ignored. The Coriolis force accompanied by the radial movement of the particles causes the droplets to bend their trajectory to the opposite direction of rotation, which can cause a collision to the subsequent blades. If the drag force that prevents this tangential motion is dominantly larger than the Coriolis force, it exits the impeller with vapour before colliding with subsequent blades. The ratio of these two opposing forces is defined as a dimensionless traceability factor:
\[ \text{Tr} = \frac{\text{Fluid force}}{\text{Coriolis force}} = \frac{0.5 \rho_p U^2 C_D S_p}{\rho_p V_p (2U \omega)} = \frac{9 \mu}{\omega \rho_p D_p^2} \]  
where \( S_p \) and \( V_p \) are the projected areas and volumes of droplets with a diameter of \( D_p \). The relative velocity of particles to vapor is small, Stokes equation was used for the drag coefficient.

\[ C_D = \frac{24 \mu}{\rho_p U D_p} \]  

It is necessary to atomize the droplet to at least \( D_p \) or less so that \( \text{Tr} > 1 \) in order to avoid collision with the subsequent blade. Figure 4 shows the relationship between the followability \( \text{Tr} \) and the droplet diameter \( D_p \), and the thick blue line is the designed rotational speed of the compressor in this paper. \( \text{Tr} > 1 \) is satisfied when \( D_p < 8 \) \( \mu \)m. The trajectories of droplets of various diameters were calculated using a three-dimensional CFD with an actual rotating impeller shape as shown in figure 5. The analysis was performed with ANSYS Fluent with discrete phase model (DPM) for droplets of various diameters. Tetrahedral mesh of 2,684,072 cells and SST k-\( \omega \) turbulence model were used. Fluid-particle interaction was only considered in the one-way mode. The particles that has entered from the inlet, did not collide to the blade if the particle size was less than 5 \( \mu \)m, where \( \text{Tr} = 3 \) as obtained from figure 4. This value can be used as a lower limit of followability to avoid the collision. The condition required for complete evaporation of the droplet is \( D_p < 10 \) \( \mu \)m [8], which is also satisfied in this case. Figure 4 also shows the case of other rotation speeds. For example, when a rotational speed is 100,000 rpm and the particle size of \( D_p > 2.4 \) \( \mu \)m, there is a high possibility of collision because the Coriolis force becomes dominant.

![Figure 4. Traceability factor and rotational speed](image)

![Figure 5. Particle trajectory analysis in CFD.](image)
3. Continuous Cooling Compressor

Table 1 shows the specifications of the turbo compressor to demonstrate CCC process. Compared with the previous study, the operating conditions and capacity are the same and the stage number and specific speed were increased. The cross-sectional view of the compressor is shown in figure 6.

Table 1. The design condition of turbo compressor

| Refrigerant | Tap water |
|-------------|-----------|
| Capacity, kW | 100       |
| Pressure ratio | 5.3     |
| Suction pressure, kPa | 0.9     |
| Suction temp. | 5.4 °C    |
| Design speed | 24500 rpm |

| Stage |
|-------|
| 1st  |
| 2nd  |
| 3rd  |

| Pressure ratio | 2.0 | 1.9 | 1.6 |
| Specific speed | 0.8 | 0.7 | 0.7 |
| Volumetric flow, m³/s | 5.7 | 3.6 | 2.4 |
| Reynolds number | 2.3×10⁴ | 4.6×10⁴ | 7.6×10⁴ |
| Isentropic efficiency | 76% | 76% | 78% |

3.1. Aerodynamics

Aiming to reduce the overall size of the compressor, specific speed of each impeller was increased by using three-stage impellers of low pressure ratios. As a result, the maximum tip speed decreased to 384 m/s from 530 m/s. As the maximum stress was also decreased, A2024-T6 material was used for the 2nd and 3rd impellers. Since the volumetric flow rate is lower for downstream stage, the pressure ratio was designed to be the highest in the first stage in order to keep the specific speed below 0.8, which is desirable value for high adiabatic efficiency. As a characteristic of water refrigerant, the Reynolds number is the order of 10⁴. This value is very low for a general turbo compressor, so care must be taken to suppress loss due to the secondary flow vortex. As a result, the blade shape with an adiabatic efficiency of 76% was obtained.

3.2. Rotordynamics and bearing

The shaft was designed as a rigid shaft, supported by saturated water lubricated bearings, to which liquid water was supplied as a lubricant. Since the three impellers are facing the same direction, the thrust forces of do not cancel each other. Therefore, an integrated bearing shown in figure 7 was developed, which has a thrust bearing surface adjacent to the end of a radial bearing. It is located between the 1st-stage and the 2nd-stage impeller as shown in figure 6. The stepped portion between the radial bearing part with a diameter of 50 mm and the cylindrical part to shrink-fit the 2nd impeller with a diameter of 100 mm was used as a thrust support surface, instead it does not have a thrust collar, which is generally used in turbo compressors for high-pressure working fluids. Such a configuration was
possible because the diameter of the central part of the shaft is large to increase the critical speed of the shaft. The bearing loss was estimated to be 2 kW from Petroff’s equation, which was twice of the tapered bearing in the previous reports, being a disadvantage of the three-stage configuration.

As shown in figure 2, centrifugal spray nozzles are located at equal intervals between all blades. The spray flow rate is changed with the number of nozzles, which is adjusted by selecting whether a nozzle or a plug is attached to the tip of the flow path. In order to reduce the rotor unbalances, a symmetric arrangement of nozzle is necessary. Therefore, there are restrictions on the number of nozzles and blades. Given that the number of blades is B, the number of nozzles must be “a divisor of B” or “B minus a divisor of B”. For example, B = 15, then the allowable numbers are 0, 3, 5, 10, 12, and 15. Thus, the number of blades should not be a prime numbers, in which the nozzles must be located between all the blades or nothing. The impeller and assembled rotor were dynamically balanced each time when the nozzle was replaced to meet the balance grade G2.5.

3.3. Spraying nozzle

Compactness and strength are required for a nozzle that is embedded into an impeller and rotates together with the impeller, as well as atomization performance. Figure 8 shows the structure of the centrifugal pressure nozzle. The fastening thread portion has sufficient strength to withstand the pulling force acting to the nozzle due to the centrifugal effect of high speed rotation. The main functional components are the swirl chamber that produces the swirl flow and the orifice in its centre. The water pressurised by the centrifugal force flows through the filter into the chamber along the tangent line of the spiral chamber. Finally, it flows into the orifice in the centre while increasing the swirling speed. Inside the orifice, a thin swirling liquid film is formed on the outer wall of the hole, and injected to the flow channel of the impeller. The generated liquid film further divides to produce fine particles.
Figure 8. Centrifugal nozzle embedded in the impellers

4. Experiment

4.1. Test rig

To demonstrate CCC, a water-refrigerant heat pump system, which was comprised of a compressor, a spray-type evaporator, a condensing ejector, and an economiser tank, was constructed. Figure 9 shows a photograph of the demonstration device and a cycle diagram. It was a three-stage compression three-stage expansion economizer cycle. When it was operated at 24,500 rpm, the pressure ratios of 1st, 2nd, and 3rd stage were obtained as 2.07, 1.85, 1.63, respectively, and 5.6 in total. Those values agreed with the designed conditions as shown in Table 1.

Figure 9. Demonstration test rig.

4.2. Discharge temperature

The discharge superheat reaches 200 K when compressed without any nozzles and without cooling, assuming the impeller efficiency of 76%. In order to investigate the continuous cooling effect of centrifugal spraying, the amount of spraying was varied by changing the number of nozzles attached to the impeller. Figure 10 shows the relation between the total spray flow rate of all stages and the discharge superheat of 3rd stage. Initially, it was estimated that the discharged vapour could be cooled to the saturation, i.e., discharge superheat of 0 K with spray flow rate of 4.4 g/s; however, the superheat of 55 K was left uncooled in the experiment. The cooling effect of 145K was confirmed, but it was insufficient. It is considered that this is because droplets larger than the Sauter average diameter do not contribute to cooling, and the adiabatic efficiency of the impeller is decreased from the analytical value. Therefore, additional experiments were performed with increased spray flow rate by increasing the number of
The discharge superheat decreased as the spray flow rate was increased, and the superheat became zero when it was 9.5 g/s.

4.3. Performance
When the spray flow rate was 4.4 g/s with the discharge superheat of 55 K, the COP had a value of 4.5. When the superheat was reduced by increasing the number of nozzle, the COP increased, reaching a peak COP value of 4.86 when the discharge superheat was 43 K. This value is lower than 5.0 that was obtained with the 2-stage intercooled compressor in the previous study; however, the increase in the efficiency by decreasing the discharge superheat with CCC was demonstrated.

When the spray flow rate was increased further, the discharge superheat decreased but the COP also decreased. In this case, the increase in the cooling capacity and pressure ratio was very small even if the compressor speed was increased. These results inferred that the impeller was choking, due to the increase in the flow rate by the existence of sprayed mist and its evaporated vapour. Although the mist was sprayed downstream of the throat, when the flow blockage due to the droplets and increase in volumetric flow is significant, the actual location of throat, at which the area is the smallest for the flow rate, would exist at downstream of the geometrical throat. From these results, it was found that there is an optimum point of CCC, and it is important to take a sufficient choke margin by increasing the exit area of the impeller of the continuous cooling compressor.

4.4. Durability
Erosion was observed on the leading edge of the blade of the impeller. This is due to a minor problem at the early stage of the experiments. It did not progress further after the problem was solved. Furthermore, no erosion was observed on the blade located downstream from the spray nozzle. Thus, the anti-erosion effect of the small relative velocity between the droplet and the blade by embedding the nozzle in the impeller was confirmed. As for the water-lubricated bearings, progressive wear was not observed, and long-term reliability owing to non-contact lubrication was confirmed.

5. Conclusion
A continuous cooling turbo compressor was proposed that sprays water droplets from the nozzle embedded in the impeller in order to reduce the superheat of discharged refrigerant water. The effect was demonstrated by experiments, and the conclusions are as follows:
1. Droplets sprayed between the blades of the impeller cool the vapour reducing the discharge superheat. As the spray flow rate was increased, the discharge superheat of the vapour decreased to the saturation. The COP was improved, given the spray flow rate was appropriate.
2. No erosion was observed at the blades due to collision of centrifugal spray droplets.
3. There is an optimum spray flow rate, at which the COP took its maximum value. When the spray flow rate is increased further, a choking at the impeller decreases the COP. This is due to the increase in the volumetric flow rate by the evaporated droplets. It is desirable to assume a sufficient choke margin or to estimate the increase in the volumetric flow rate using a special CFD code that can consider the divergence due to droplet evaporation.

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