Experimental Study on Compact Heat Pump System for Clothes Drying Using CO₂ as a Refrigerant

M. Honma, T. Tamura, Y. Yakumaru and F. Nishiwaki
Matsushita Electric Industrial Co., Ltd.
Living Environment Development Center
3-1-1, Yagumo-Nakamachi, Moriguchi, Osaka, 570-8501, Japan
Phone:+81-6-6906-2821, Fax:+81-6-6906-2865, E-mail: honma.masaya@jp.panasonic.com

Abstract

This paper presents an experimental study on a compact heat pump system for drying using CO₂ as the refrigerant.

As one of the countermeasures against global warming and energy conservation problems, natural refrigerants such as CO₂ are now being investigated as substitutes for HFCs for the supply of hot water mainly. CO₂ refrigerant with heating performance superior to that of HFC refrigerants can be adopted even for heating uses other than supplying hot water.

In this study, the following measures were taken for the construction of a compact heat pump system for drying using CO₂ as a refrigerant.

1) High performance devices such as a gas cooler, evaporator and compressor were developed to provide a compact heat pump drying system using the CO₂ trans-critical cycle.
2) The CO₂ heat pump package was designed to provide optimum air flow distribution passing through the heat exchangers.
3) An optimum heat pump cycle control method was developed to improve the COP and shorten the drying time, for example, an optimum super heat control using an expansion valve.

The prototype of CO₂ heat pump dryer system was successfully developed. The drying performance of this system was far superior to that of current heater systems. Electric power consumption can be reduced by 59.2%, and the drying time can be reduced by 52.5% in comparison with heater dryer system. Furthermore the drying time can be reduced approx. 3% by controlling superheat to a constant 6-10deg.

1. Introduction

In recent years, environmental problems such as the depletion of the ozone layer and global warming have become evident and the regulation of emissions of greenhouse gases is being promoted. With this as a background, in the sphere of refrigeration and air-conditioning, natural refrigerants which have little impact on the environment have become the center of attention. In particular, carbon dioxide (CO₂) refrigerant, which has an ozone depletion potential of zero, a global warming potential of 1 and which is nonflammable and nontoxic, has come to be considered as one substitute for HFCs. A CO₂ heat pump water heater making use of the outstanding heating characteristics of CO₂ refrigerant has been released (trademark: Eco-Cute) and great hopes are placed on further expansion in the market for CO₂ heat pump systems.
This paper investigates the possibilities of applying such a CO₂ heat pump to a clothes washer/dryer system. Conventional clothes dryers heat the air used for drying with an electric heater and heat and dry the clothes by having the heated air come into direct contact with them. Because of this and the fact that they apply a water-cooled dehumidification method, using water as coolant to remove the moisture from the highly humid air, they have a very high electric power consumption and use a lot of water when drying. In contrast to this, heat pump dryers apply a drying method which uses a highly efficient heat pump cycle gas cooler in heating and an evaporator in dehumidifying. This means that they do not require a heater or water for drying, enabling power consumption and water usage to be greatly reduced in comparison with conventional methods.

In this research, we develop a compact high-efficiency CO₂ heat pump system, which was able to be installed in domestic washer/dryers, and compared its performance with conventional dryer systems.

2. Heat Pump System for Drying

2.1 Heat Pump System

A diagram of a heat pump clothes dryer equipped with a heat pump is shown in Fig. 1.

![Diagram of Heat Pump System](image)

The refrigerant in Fig. 1 circulates through the system in the direction shown by the arrows, where it is compressed to high pressure and high temperature in the compressor, radiates heat to the air used for drying in the gas cooler, expands to reduced pressure in the expansion valve, absorbs heat and evaporates and then returns to the compressor. The air used for drying circulates through the duct in the direction shown by the dotted arrows, during which process it absorbs moisture from the clothes, is dehumidified and cooled in the evaporator, heated in the gas cooler and then is returned again to the drum as dry warm air. When the heat pump is installed in a clothes washer/dryer, because the air circulation route is a closed cycle, as opposed to the open cycle in an air conditioner or similar, it is necessary to carry out adjustment of the balance between the amount of heat applied to the air and the amount lost by radiator, control in regard to changing the refrigerant cycle according to how drying is progressing, etc.
2.2 Refrigerant Characteristics of CO₂

When setting out to develop the heat pump dryer, first, the most suitable refrigerant was selected. As refrigerants, we compared CO₂, as used in the heat pump water heater, and HFC134a, as used in refrigerators and car air conditioners. Table 1 shows the main refrigerant characteristics of CO₂ in comparison with HFC134a. The Global Warming Potential (GWP) of CO₂ refrigerant is 1/1300 compared with that of HFC134a and the direct effect on the environment is extremely small. The temperature distribution in refrigerant and air when heating air in a gas cooler using HFC134a and CO₂ are shown in Figs. 2(a) and 2(b). CO₂ refrigerant possesses the feature that the refrigeration cycle becomes a trans-critical cycle. As can be seen in Fig. 2(b), in the supercritical state there is no condensation process causing a vapor-liquid state, so the temperature of refrigerant in the gas cooler changes constantly. Therefore, in comparison with HFCs in which the condensation process is evident, when CO₂ refrigerant is used, it is possible to reduce the temperature difference between the refrigerant in the gas cooler and the fluid being heated. CO₂ refrigerant also proves to have an advantage in being able to increase the temperature of the fluid being heated to higher temperatures. For these reasons, by using CO₂ refrigerant, it becomes possible to carry out high temperature drying not feasible with HFC134a.

<table>
<thead>
<tr>
<th></th>
<th>HFC134a</th>
<th>CO₂</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ozone Depleting Potential (ODP)</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Global Warming Potential (GWP)</td>
<td>1300</td>
<td>1</td>
</tr>
<tr>
<td>Critical Temperature [℃] / Critical Pressure [MPa]</td>
<td>101.1/4.06</td>
<td>31.1/7.38</td>
</tr>
</tbody>
</table>

Table 1 Comparison of refrigerant characteristics

![Fig.2 Temperature distribution of refrigerant and heated fluid](image)

3. Development of CO₂ Heat Pump System for Highly Efficiency

3.1 Forecast of Heat Pump System Performance

First, we predicted the performance characteristics of the heat pump dryer system. The conditions used for analysis are shown in Table 2. We defined that heating capacity was heating capacity of gas...
cooler, and that the water which evaporated from clothes dehumidified all at the evaporator, and that the differences between heating capacity of gas cooler and cooling capacity of evaporator (equal in compressor input) were released from the duct. The weight of clothes (dry condition) was 4.5kg.

<table>
<thead>
<tr>
<th>Table2</th>
<th>Design conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heating capacity of gas cooler</td>
<td>kW</td>
</tr>
<tr>
<td>Minimum temperature difference between water and refrigerant</td>
<td>deg</td>
</tr>
<tr>
<td>GC inlet air temperature</td>
<td></td>
</tr>
<tr>
<td>GC outlet air temperature</td>
<td></td>
</tr>
<tr>
<td>GC inlet refrigerant temperature</td>
<td></td>
</tr>
<tr>
<td>Refrigerant saturated temperature in evaporator</td>
<td></td>
</tr>
<tr>
<td>Compressor efficiency</td>
<td>%</td>
</tr>
</tbody>
</table>

The heat balance between the air side and refrigerant cycle side was analyzed using a refrigerant cycle simulation. Compressor efficiency was taken to be 60%. The results of analysis of the temperature distribution between refrigerant and air are shown in Fig. 3 and the Mollier diagram for the heat pump dryer system is shown in Fig. 4. From these results, coefficient of performance (heating capacity of gas cooler divided by compressor input, referred to below as COP) was estimated as 4.07.

![Fig.3](image-url) Predicted temperature distribution of refrigerant and air

![Fig.4](image-url) Calculation result of drying cycle

**3.2 Optimization of Heat Exchanger**

Next, we developed high efficient components for the heat pump cycle. Fin tube heat exchangers were adopted in the gas cooler and evaporator and an electric expansion valve with a controllable aperture was adopted for the expansion device.

First, a study was carried on the specifications for the heat exchanger. In terms of the specifications for the dryer, a warm air temperature of 80°C, a fan air volumetric flow rate of 2.4 m³/min. and pressure drop on the air side of 120 Pa or less were established and a prediction was made for the specifications of a heat exchanger satisfying these conditions using a simple
simulation. The conditions for the analysis were: Front face of heat exchanger: H220 x D140 [mm];
External diameter of tube 5.0 (thickness 0.27) [mm]; Front face air velocity U = 1.0 m/s; Inlet air
temperature: 18°C; Inlet refrigerant temperature: 102°C.

The method of calculation was a block subdivision method\(^1\) in order to take consideration of the
refrigerant circuit pattern. The amount of heat exchanged in each block Q was calculated using the
following equation:

\[
Q = G_a \cdot C_{pa} \cdot \Phi (T_a - T_r)
\]  

(1)

Here, \(\Phi\) is heat exchanger effectiveness, calculated according to the equations below:

\[
\Phi = 1 - \exp[-KA_a/(G_a \cdot C_{pa})]
\]  

(2)

\[
1/K = 1/\alpha_a - (A_a/A_r)/(1/\alpha_r)
\]  

(3)

\(\alpha_a\) is the overall heat transfer coefficient on the air side, including contact resistance and fin
efficiency and is predicted using the Wilson plot method. \(\alpha_r\) is the heat transfer coefficient on the
refrigerant side, calculated according to the following equation:

\[
\alpha_r = C \cdot \alpha_{ra}
\]  

(4)

Here, \(\alpha_{ra}\) is the heat transfer coefficient of a single component refrigerant in a horizontal smooth
pipe and is calculated using the method in Dang et al.\(^2\) \(C\) is a correction factor, calculated through
experimentation.

Studies was made on optimal compactness and optimal efficiency from the viewpoint of
increasing the counter flow effect by increasing the rows in regard to the air flow to the maximum
extent in order to make use of the characteristic of CO\(_2\) that there is no condensation process
causing a vapor-liquid state, allowing the temperature of refrigerant in the gas cooler to change
constantly.

From the results of simulation, as shown in Fig. 5, it becomes apparent that 10 rows in the heat
exchanger are sufficient to keep air pressure drop to 120 Pa and under, while maintaining a heat
exchange of 2.7 kW. A similar simulation was carried out for the evaporator and 6 rows decided
upon. The number of passes was set at 1 pass for the gas cooler and 3 passes for the evaporator
from the viewpoint of refrigerant pressure drop in the heat exchanger and cycle performance. In
addition, to install the heat pump in the space available in the current washer/dryer model (W400 x
H250 x D215), the gas cooler was designed at H220 mm x D140 mm and the evaporator at H230
mm x D150 mm.

We then developed the duct carrying the air flow for drying that the gas cooler, evaporator and compressor are housed in. The heat exchangers (for gas cooler and evaporator), the compressor, distributor and expansion valve were housed inside the duct for compactness and the heat pump system was constructed so that radiant heat from the compressor could be used for drying, as shown in Fig. 6. In addition, using fluids analysis software (Fluent), the duct was designed so that...
air flow distribution would be uniform over the front face of the heat exchangers. Analysis was carried out using a k-ε turbulence model. As a result, a duct shape was established allowing an almost uniform air flow distribution, as shown in Fig. 7. In creating such an uniform air flow distribution, the system COP is improved by 11.6%.

![Fig.6 Model of fluid analysis](image1)

![Fig.7 Result of fluid analysis (Front view)](image2)

### 3.3 Compact Highly Efficient CO2 Compressor

We then developed a compact highly efficient CO2 rotary compressor which was able to be installed in the air duct that houses the heat pump system shown in Fig. 6. For the purposes of developing a highly efficient heat pump system, we concentrated on minimizing the lubricating oil discharged from the compressor to the heat pump system. In general, if the oil circulating ratio increases in the heat pump system, the oil resists heat transfer in the heat exchanger or leads to an increase in refrigerant pressure drop, worsening the performance of the heat pump cycle. The main points of these measures are: (1) Reduction of oil inflow to the compression chamber. (2) Promotion of oil separation in the high pressure vessel. For (1), by attempting optimization of the seal length of the top and bottom faces of the pistons and the seal length of the vane grooves, the oil circulating ratio (the ratio of the oil mass flow rate in circulation in regard to the overall mass flow rate in circulation) was reduced to 0.4 wt%. For (2), the results of analysis of flow fields in the high pressure vessel using multiphase flow analysis indicated that oil separation was being impeded by swirling flow caused by the rotation of the rotor. To solve this problem, baffles and a journal bearing cover was installed, allowing an oil discharge amount of 0.08 wt% to be achieved. As a result, the system COP was improved by about 5%. In addition, the cylinder volume of the newly developed rotary compressor is 1.32 cc, the shell diameter is Φ88 mm with a height of 175 mm.

### 3.4 Optimization of Control Methods

We then carried out an investigation of optimal control methods for the purpose of realizing high efficiency and shortening drying time. In the drying process, the state of the circulating air used for drying is constantly changing from a very humid state when drying first starts to a state of low humidity in the later stages of drying. Therefore, it can be considered that a high efficiency heat pump system can be realized by controlling the expansion valve to the optimum setting depending on the state of the circulating air at any one time. When the expansion valve aperture is fixed, the temperature of the air blown from the gas cooler decreases with time, as shown in Fig. 8.

This results from a decrease in the compressor discharge temperature due to a decrease in the
high pressure and super heat, which causes a decrease in heating capacity. To solve this problem, we investigated controlling the expansion valve so that super heat in the drying process is kept constant. Fig. 9 compares heating capacity when super heat is controlled to stay in the 4-8 deg and 6-10 deg range, and when the expansion valve aperture is constant (no super heat control). It is clear that the decrease in heating capacity can be suppressed by controlling super heat to a constant level. It is also clear that the target heating capacity of 2.7 kW can be maintained by controlling super heat to 6-10 deg. In addition, by controlling super heat to 6-10 deg, drying time can be reduced by 3%.

Fig. 9 Heating capacity with and without SH control

4. Prototype System

A prototype of the compact CO₂ heat pump for installation in domestic clothes washer/dryers is shown in Fig. 10. Fig. 10 is a front view. The evaporator, gas cooler and compressor are installed in sequence in the duct, in the direction of the air flow.

The specifications of the prototype are shown in Table 3. The results of a performance evaluation of the prototype system, as shown in Table 4, give an electric power consumption of 1,142 Wh (a reduction of 59.2% in comparison with heater types), a drying time of 95 minutes (a reduction of 52.5% in comparison with heater types), water usage for cooling during drying as zero (50 L with heater types) and a COP of 3.76. The water of 1.5kg was evaporated from the clothes, and the water was dehumidified at the evaporator. Measured operating points for the refrigerant cycle are shown in Fig. 11. When these

<table>
<thead>
<tr>
<th>Table 3 Specifications of prototype system</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Compressor</strong></td>
</tr>
<tr>
<td><strong>Gas cooler</strong></td>
</tr>
<tr>
<td><strong>Evaporator</strong></td>
</tr>
<tr>
<td><strong>Distributor</strong></td>
</tr>
<tr>
<td><strong>Fan</strong></td>
</tr>
</tbody>
</table>
operating points are compared to the assumed operating points shown in Fig. 4, the following can be considered causes of a decrease in COP: (1) Compressor efficiency was less than the assumed value of 60%; (2) Air flow resistance was increased due to condensed water in the evaporator; (3) Flow distribution of refrigerant in the evaporator is uneven. In the future we plan to tackle such problems as these for even higher performance.

Table 4  Results of CO2 heat pump system performance compared with heater system

<table>
<thead>
<tr>
<th></th>
<th>Heater type</th>
<th>CO2 heat pump type</th>
</tr>
</thead>
<tbody>
<tr>
<td>Electric power consumption [Wh]</td>
<td>2800</td>
<td>1142</td>
</tr>
<tr>
<td>Drying time [min]</td>
<td>200</td>
<td>95</td>
</tr>
<tr>
<td>Water consumption for drying [L]</td>
<td>50</td>
<td>0</td>
</tr>
<tr>
<td>COP[-]</td>
<td>1</td>
<td>3.76</td>
</tr>
</tbody>
</table>

5. Conclusion

We developed a highly efficient compact CO2 heat pump dryer system for installation in domestic clothes washer/dryers and obtained the following results:

1) We constructed a CO2 heat pump prototype system which was able to reduce electric power consumption by 59.2% and drying time by 52.5% in comparison with heater dryer systems.
2) We established an optimum super heat control method which was able to avoid a reduction in heating capacity and reduce drying time by approx. 3% by controlling super heat to a constant 6-10 deg.

6. References