# DEVELOPMENT OF TWO-PHASE FLOW EXPANDER FOR $CO_2$ HEAT PUMPS AND AIR-CONDITIONERS

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## ABSTRACT

We developed a two-phase flow expander for CO2 heat pumps and air-conditioners to improve the performance of transcritical cycle. We selected the swing type as an evaluation result of the existing pump and compressor types from the viewpoints of performance, reliability with two-phase flow, applicable volume, and productivity when applied to expanders. The test results of the newly designed prototype of a two-stage swing expander showed high efficiency in energy recovery. According to the evaluation made by calculation when mounted on a heat pump, the expander gives substantial reduction in heating energy consumption when compared with traditional combustion type heaters.

**Key Words:** ***heat pumps, $CO_2$, expander***

## 1 INTRODUCTION

Air-conditioning systems, which provide both cooling and heating mode applied in the moderate climate regions in Japan, adopt the HFC refrigerant vapor compression heat pumps due to its high energy efficiency. However, in comparison with electric heaters or combustion type heaters like furnaces, the heat pump efficiency falls when the discharge air temperature is raised high and the environment becomes less comfortable. Therefore, heating by heat pumps are not widely spread in cold regions. If heat pumps that can provide comfortable heating are developed, we can expect more reduction in energy consumption. In terms of primary energy efficiency, heat pumps give more than 120% while the combustion type gives only 80 to 85%.

Heat pumps with CO2 give lower Coefficient of Performance (COP) than those with HFC when used under the condition of normal vapor compression cycle. However, the characteristics of CO2 enables to produce higher temperature discharge air than HFC with a relatively small capacity drop even at a low outdoor temperature, which results in creating a more comfortable environment. If the COP of the CO2 heat pumps can be raised at least equivalent to that of HFC, heat pumps that provide comfort in cold regions and yet reduce primary energy consumption are obtainable. In addition, if the HFC refrigerants are replaced with the natural refrigerant CO2, it will help to reduce emissions of green house gases.

## 2 EXPANDER CYCLE

Out of many schemes proposed for improvement in the CO2 refrigerating cycle performance, the most effective way is to apply the expander cycle. In general, as shown in Fig. 1, when the refrigerant cooled by a gas cooler is expanded through a throttle valve from high to low pressure, theoretically the process is iso-enthalpic. If the throttle valve is replaced with an expander, it recovers the energy created in the iso-entropic expansion process. If the recovered energy is utilized for driving the compressor, it reduces the power consumption of a refrigerating cycle.

## 3 SELECTION OF EXPANDER TYPE

First, we evaluated on paper the existing pumps and compressors by type from the viewpoint of their applicability to expanders. The evaluated items are performance, reliability, applicable volume, productivity et al. Above all, the first priority was given to efficiency. The second priority was given to mechanical reliability, because the CO2 refrigerant flow through an expander is in two-phase and its working pressure is considerably high.

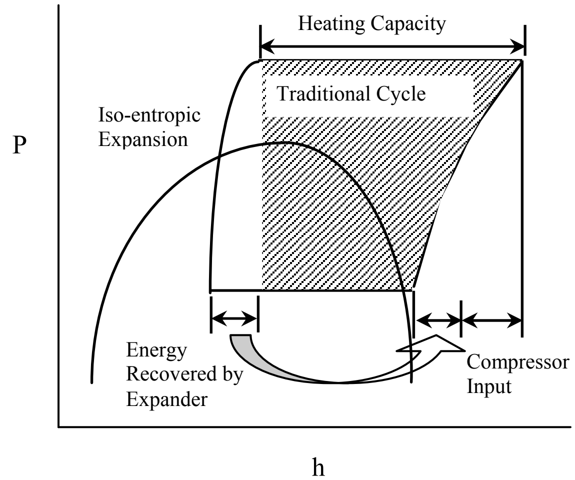



**Fig. 1. Principle of expander cycle**

From the evaluated results shown in table 1, we selected the swing type as the first candidate and the scroll type as the second candidate for the expander. The reliability of the swing type is superior to that of other types due to its superior sliding characteristics and the mechanical internal construction with least leakage. If the swing type is designed properly for effective expansion, the energy recovery efficiency can be improved. We at Daikin Industries are currently producing the swing type compressors in quantity and have accumulated technologies of swing type. This is another good reason for selecting the swing type as the first candidate.

**Table 1. Comparison of types for expander**

| Base | Compressor | | | | | Vacuum Pump | Pump / Motor | Turbine |
|---|---|---|---|---|---|---|---|---|
| Type | Scroll | Screw | Recipro-cating | Rolling Piston | **Swing** | Roots | Gear | Centrifugal |
| Performance | Good to Average | Average to Bad | Good to Average | Average | Good to Average | Average | Average | Bad |
| Torque Fluctuation | Good | Good | Bad | Average | Average | Good to Average | Good | Good |
| Noise | Good | Average | Average | Good | Good | Average | Average to Bad | Good |
| Reliability | Average to Low | High to Low | Average to Low | Low | High | Average | Average to Low | High |
| Applicable Volume | Middle | Large | Small to Middle | Small to Middle | Small to Middle | Small to Middle | Small | Large |
| Structure Complexity | Average | Average to Complex | Average to Complex | Simple | Simple | Simple | Simple | Simple |
| Number of Shafts | 1 | 2 | 1 | 1 | 1 | 2 | 2 | 1 |
| Connection with Compressor | Average | Average | Easy to Difficult | Easy | Easy | Average | Average | Easy |
| Productivity | Average | Average to Bad | Good | Good | Good | Good | Good | Good |
| Comprehensive Evaluation | Good to Average | Average | Average | Average to Bad | Good | Average | Average | Bad |

## 3 SWING EXPANDER

### 3.1 Basic Structure

The swing type expander must have a mechanism to control the flow rate of incoming high pressure refrigerant. We propose the two-stage expansion as shown in Fig. 2. Its structure is similar to the two-stage compressor except that there is no valve between the two cylinders.

The expansion process is shown in Fig. 3. The high-pressure side of the 1st cylinder becomes an inlet chamber. The low-pressure side of the 1st cylinder and the high-pressure side of the 2nd cylinder become an expansion chamber. The low-pressure side of the 2nd cylinder becomes an outlet chamber. The volume of the 2nd cylinder is designed to be larger than the 1st cylinder, and as the piston rotates, the volume of the expansion chamber increases because the volume of the expansion chamber of the 2nd cylinder increases more than the volume of the expansion chamber of the 1st cylinder decreases.

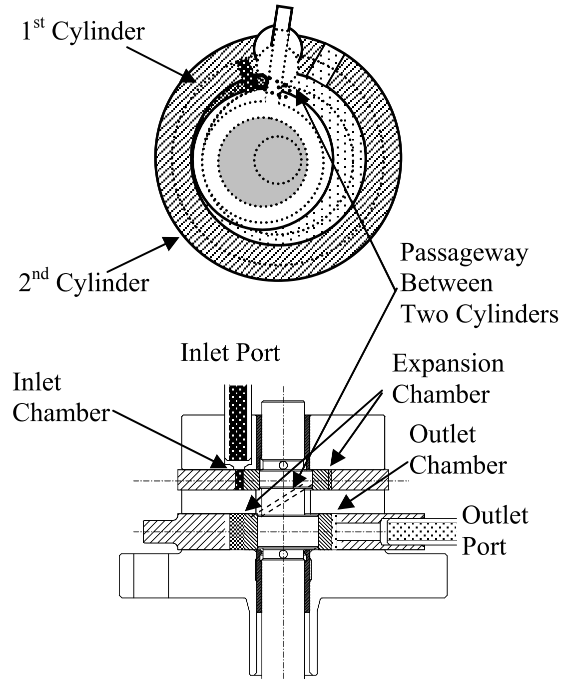



**Fig. 2. Structure of two-stage expander**

Thus, by separating each chamber of inlet, expansion and outlet, the refrigerant flow during one rotation becomes continuous, which results in reduction of the suction pulsation caused by shutting off the refrigerant flow.

### 3.2 Testing Device

Figure 4 shows the schematic of the device for tetsting the expander performance. The temperatures(T), pressures(P), and mass flow rates(G) are measured at the designated points. In addition,

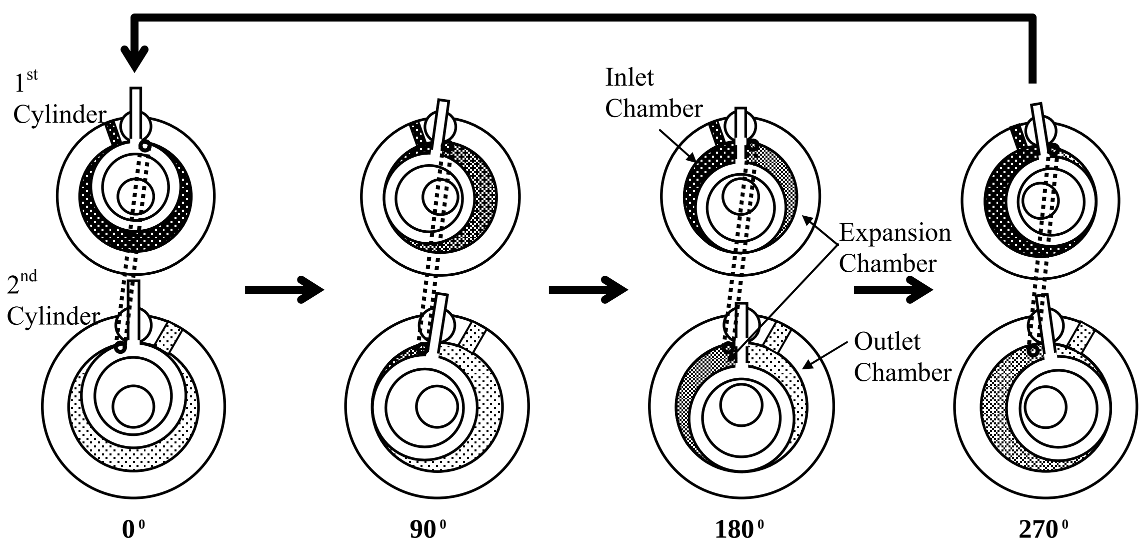



**Fig. 3. Expansion process of two-stage expander**

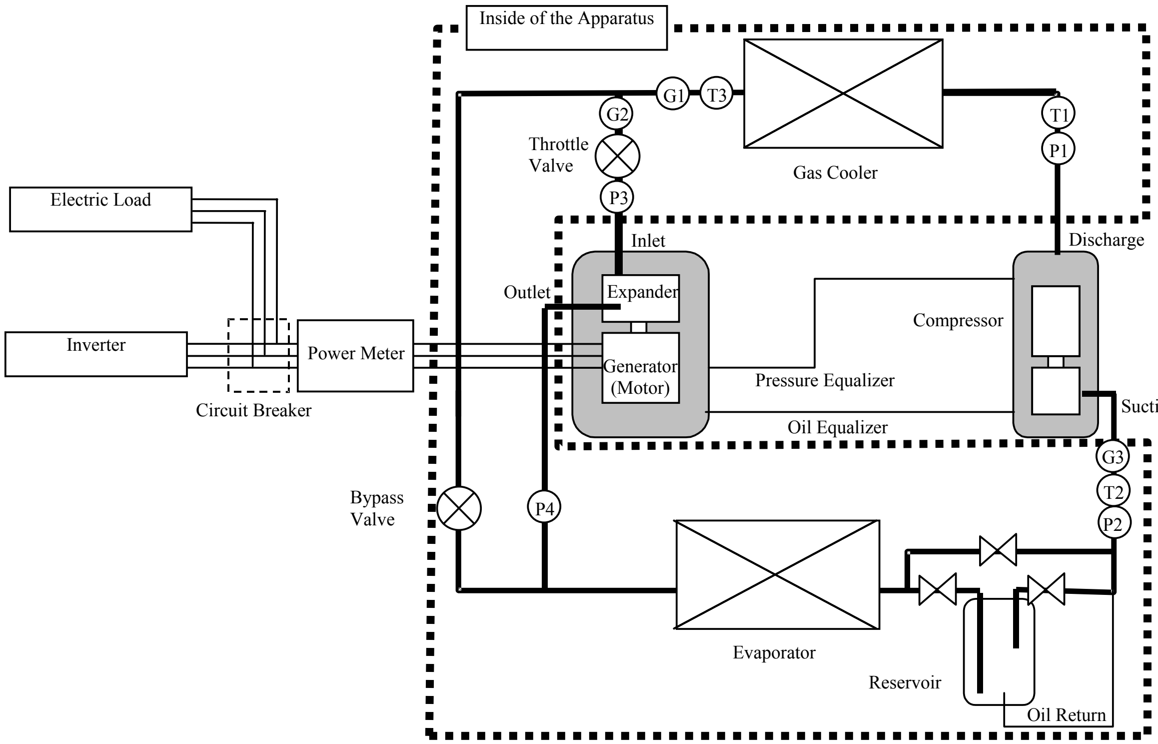



**Fig. 4. Schematic of testing device**

the radiated heat from the gas cooler is measured by the flow rate and temperature difference of the cooling water and the input to the evaporator is measured by the input to the electric heater. The expander and the compressor unit can be independently set up on the testing device as in Fig. 4 as well as the expander-compressor unit described later. When the expander is independently set up on the testing device, it is connected with the compressor with a pressure equalizer and an oil equalizer. The expander is connected with a motor (generator) and the rotation is initiated by the inverter controlled motor. After the expander starts rotating by itself, the circuit breaker is switched to the electric load to control the rotation speed by controlling the load. The power meter measures the recovered energy.

The expander efficiency $\eta_{\exp}$ is described as the ratio of the recovered energy $W_{rec}$ to the theoretically recoverable energy $W_{th}$ (Eq. 1).

$$\eta_{\exp} = \frac{W_{rec}}{W_{th}} = \eta_{ind} \times \eta_{mech} \tag{1}$$

$W_{th}$ is calculated from $G_2$ measured by the mass flow meter 2 in Fig. 4, the inlet enthalpy $h_{in}$ calculated from $T_3$ and $P_3$ and the outlet enthalpy $h_{out}$ calculated under the condition of iso-entropic expansion from $T_3$ and $P_3$ (Eq. 2).

$$W_{th} = G_2 \times \left(h_{in} - h_{out}\right) \tag{2}$$

The recovered energy $W_{rec}$ is calculated from the energy measured by the power meter $W_{meas}$ and the generating efficiency of the generator $\eta_{gen}$ formerly evaluated (Eq. 3).

$$W_{rec} = \frac{W_{meas}}{\eta_{gen}} \tag{3}$$

The inner pressure of the expander is measured to calculate the work indicated (Eq. 4) and the efficiencies are evaluated (Eqs. 5-7).

$$W_{ind} = \sum PV \tag{4}$$

$$\eta_{ind} = \frac{W_{ind}}{W_{th}} \tag{5}$$

$$\eta_{mech} = 1 \quad \frac{W_{loss}}{W_{ind}} \tag{6}$$

$$W_{loss} = W_{rec} - W_{ind} \tag{7}$$

The volumetric efficiency of expanders is reciprocal to that of compressors because the refrigerant leaks from inlet to outlet of expanders increasing the mass flow rate (Eq. 8).

$$\eta_{vol} = \frac{V_{cyl} \times R}{G_2} \tag{8}$$

### 3.3 Test Results

The test results of the expander prototypes are shown in Table 2. The test condition is determined for the Japanese standard heating mode whose high pressure is 9.4MPa, low pressure is 3.4MPa (saturated temperature is –1.3 $^{0}$C) and the inlet temperature to the expander is 30$^{0}$C.

**Table 2. Test results of two-stage expander**

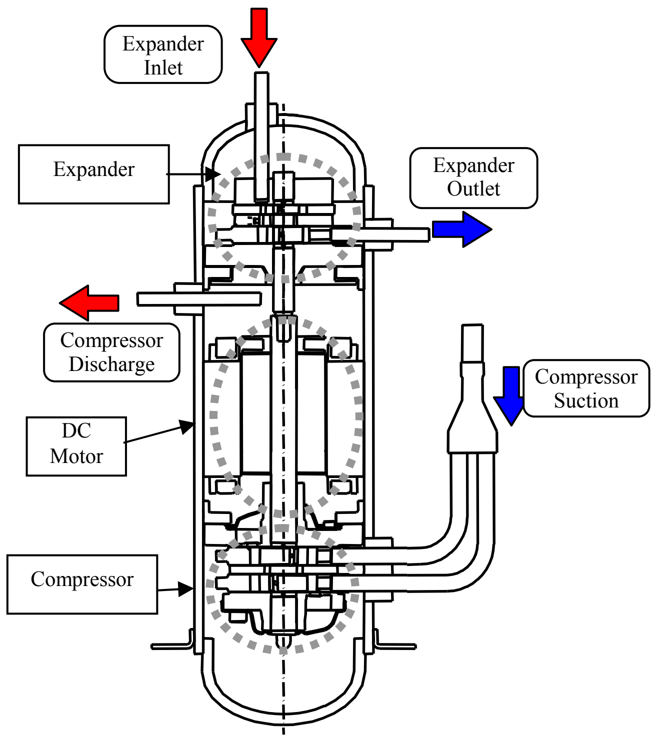
| | |
|---|---|
| Expander Efficiency $\eta_{\exp}$ | 59% |
| Volumetric Efficiency $\eta_{vol}$ | 98% |
| Indicated Efficiency $\eta_{ind}$ | 93% |
| Mechanical Efficiency $\eta_{mech}$ | 65% |

## 4 DEVELOPMENT OF EXPANDER-COMPRESSOR UNIT

There are several ways to make use of the recovered energy by the expander, but we chose to drive the compressor directly by the power recovered. For this reason, the test was conducted on an expander-compressor unit with the expander prototype of final specification.

### 4.1 Structure of Expander-Compressor Unit

The structure of the expander-compressor unit is shown in Fig. 5. The compressor, motor and the expander are connected with a power transmission shaft inside a shell sequentially from the bottom. The unit is filled with high-pressure refrigerant, and the discharged refrigerant from the compressor flows out from the outlet mounted above the motor after the motor being cooled.




(a) Cross Section

(b) Outlook

**Fig. 5. Structure of hermetic prototype of expander-compressor Unit**

### 4.2 Test Results

The test results are shown in Fig. 6. The electric power input to the compressor was reduced by 12% at 55rps, which provides a suitable heating capacity under the heating condition stipulated by the Japanese Industrial Standard.

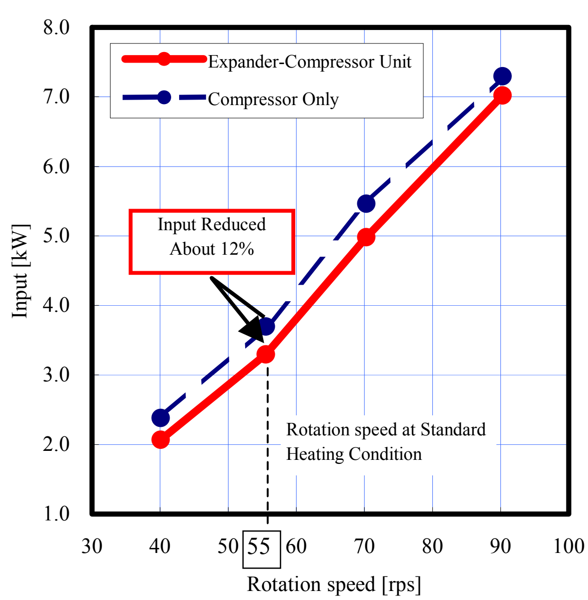



**Fig. 6. Test results of expander-compressor unit**

## 5 EVALUATION OF HEAT PUMPS WITH THE EXPANDER

The theoretical simulation based on the expander test results when applied to heat pumps resulted in the following data. The heat pump consists of one indoor unit and one outdoor unit connected by refrigerant pipes, which is the typical type in Japan.

### 5.1 Performance of the Heat Pump

The heat load in Sapporo, a representative city in Japanese cold region, and the performance of the heat pump are shown in Fig. 7 according to the outdoor temperature change. Simulation is based on the condition that the discharge air of $45^0$C was delivered into a room of $20^0$C. The input to the fan and the loss of heat and pressure in pipes and components are taken into account after the theoretical calculation of the heat pump performance. It is also taken into account that the performance drops 10% below $5^0$C because the heat pump needs a defrosting of the evaporator. According to the calculation, the heat pump performs COP 3.7 at outdoor temperature of $7^0$C, in the Japanese standard heating condition.

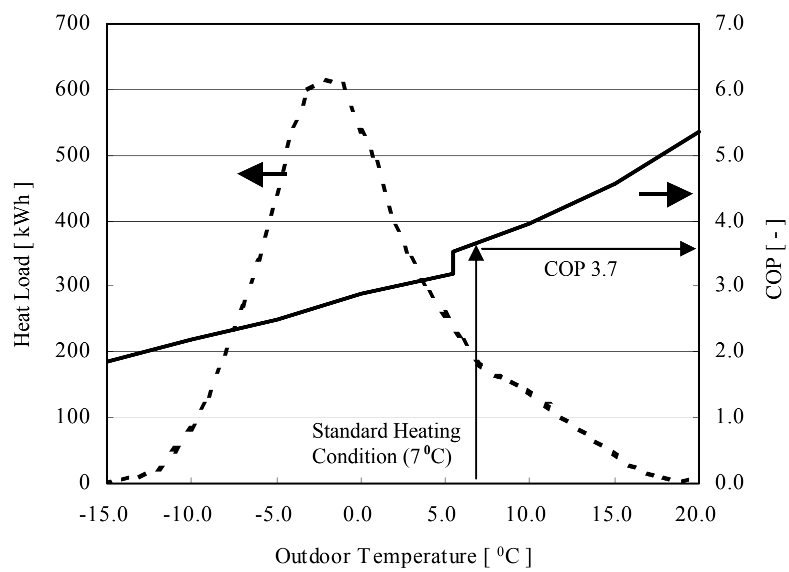



**Fig. 7. Calculate conditions of heat pump**

### 5.2 Effect on Reduction of Energy Consumption

With the results mentioned above, the heat pump's seasonal performance during heating season was evaluated. The seasonal COP is calculated as:

$$\overline{COP} = \frac{\Sigma L(T)}{\Sigma\big(L(T)/COP(T)\big)} \tag{9}$$

The result is shown in Table 3. The seasonal performance in heating is 2.85 and the primary energy efficiency is 107% with the electric power generation efficiency of 40.17% and power loss factor of transmission and distribution of 6.8% in Sapporo city (Hokkaido Electric Power co., Inc., 2003). The combustion type heaters mainly used in northern regions in Japan aims at the primary energy efficiency of 86%. Therefore, the heat pumps with expanders exceed the traditional combustion type heaters by 24% in primary energy consumption.

**Table 3. Calculated results of primary energy efficiency**

| | CO2 Heat Pump | Combustion Type Heater |
|---|---|---|
| Seasonal Performance During Heating Season | COP 2.85 | 0.86 |
| Primary energy efficiency | 107% | 86% |

## 6 CONCLUSION

(1) We focused on the expander cycle to improve the performance of CO2 heat pumps and selected the swing type for the expander.
(2) The evaluation was made with expander prototypes. The expander efficiency of 59% with the two-stage swing expander was obtained.
(3) The test results of expander-compressor prototype showed reduction of input to compressor by 12%.
(4) The above test results were applied to simulation of heat pump performance and their results are as follows: COP 3.7 is obtained in the Japanese standard heating condition (outdoor temperature $7^0$C). The primary energy efficiency during heating season in Japanese cold regions is 107%, which reduces energy consumption by 24% comparing to the traditional combustion type heaters.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## NOMENCLATURE

| | | | |
|---|---|---|---|
| G | mass flow rate | T | temperature |
| L | load | V | volume |
| P | pressure | W | work |
| R | rotation speed | | |

Greek Symbol

| | |
|---|---|
| $\eta$ | efficiency |

Subscripts

| | | | |
|---|---|---|---|
| cyl | cylinder | mech | mechanical |
| in | inlet | out | outlet |
| ind | indicated | rec | recovered |
| gen | generator | th | theoretically recoverable |
| loss | loss | vol | volumetric |
| meas | measured | | |