2008

Development of High-Efficiency Technology of Two-Stage Rotary Expander for CO2 Refrigerant

Masaru Matsui
Matsushita Electric Industrial Co.

Masanobu Wada
Matsushita Electric Industrial Co.

Takeshi Ogata
Matsushita Electric Industrial Co.

Hiroshi Hasegawa
Matsushita Electric Industrial Co.

Development of High-Efficiency Technology of Two-Stage Rotary Expander for CO2 Refrigerant

Masaru Matsui[1]*, Masanobu Wada[1], Takeshi Ogata[1], Hiroshi Hasegawa[1]

[1]Living Environment Development Center, Matsushita Electric Industrial Co., Ltd.,
3-1-1 Yagumo-naka-machi, Moriguchi City, Osaka, 570-8501, Japan
Tel: +81-6-6906-4846, Fax: +81-6-6904-5163, E-Mail: matsui.masaru@jp.panasonic.com

ABSTRACT

In order to improve the coefficient of performance (COP) of a heat pump cycle by using carbon dioxide (CO2), an expander that recovers rotation power from the large pressure difference in the cycle to use the power to partially drive the cycle was developed. In the cycle a two stage rotary expander was adopted because the suction control mechanism is not necessary in it. In the development process of pursuing higher efficiency, a performance analysis simulation model was developed by combining the dynamic mechanical analysis and the refrigerant pressure analysis in the expander by taking refrigerant leakage into consideration. After examining the impact of each design parameter on the performance by using the simulation, an optimized design has been produced and the expander efficiency of 60% along with the improvement of cycle COP by 6% have been achieved in a short term.

1. INTRODUCTION

Since The Third Session of the Conference of the Parties to the United Nations Framework Convention of Climate Change (COP3, Kyoto Conference) held in December 1997, the need for energy conservation in the household sector has become increasingly required. In Japan, especially energy conservation of heat-related home appliances has become an important issue, because approximately 50% of CO2 emissions have been generated by heating rooms (28%) and heating water (23%). In 2001, a heat pump water heater called "ECO CUTE" operating with carbon dioxide (CO2) refrigerant was introduced in the market. Due to the strong points that it uses low cost off-peak power for heating water and it can reduce CO2 emissions by using atmospheric thermal energy, the sales of this water heater have been increasing at a rate higher than 50% a year. The rated COP of "ECO CUTE" has been improving annually from 3.46 with the first model to more than 4.90 with the recent model, marking a performance improvement of 20% or higher.

The state of CO2 refrigerant in the "ECO CUTE" heat pump cycle is supercritical single phase in the high-pressure side and gas-liquid dual phase in the low-pressure side with the difference of more than 60 atmospheric pressures. The previous activities to improve COP were limited to improve the efficiency of compressors and heat exchangers and the structure of the cycle. No effective measures have ever been taken in the expansion valve even though it wastes a large amount of pressure difference energy. In order to obtain further improvement of COP, to replace the expansion valve with an expander is an effective means for recovering the drive power from the large pressure difference.

An expander can operate by reverse-rotating a compressor, but it will face its unique problems such as suction control mechanism and drastic change of CO2 refrigerant from supercritical single phase to gas-liquid two-phase in the expansion process. First of all, we chose a two-stage rotary expander because it requires no suction control mechanism, then took the approach to use analytical technologies for optimizing the design parameters of the expander in the development process. The performance analysis simulation for predicting the expander performance was developed by combining the dynamic mechanical analysis and the refrigerant pressure analysis in the expansion chamber by taking refrigerant leakage from clearances into consideration. The refrigerant pressure analysis adopted a method of updating the CO2 property in the expansion chamber at every increment by linking it with the standard reference database (REFPROP) of the National Institute of Standards and Technology (NIST). This method has yielded an accurate calculation of the CO2 refrigerant expansion process from supercritical single phase to gas-liquid two-phase.

To optimize the design parameters of two-stage rotary expander, the effect of each design parameter on the expander performance was examined by changing sets of parameters based on an orthogonal array and applying the calculation results to 'design of experiments' . By applying the development process that uses simulations to the design, prototyping, a reduction of the development lead-time was achieved.

2. STRUCTURE OF A TWO-STAGE ROTARY EXPANDER

2.1 Outline of a Compressor Combined with Expander

The developed two-stage rotary expander has been designed to connect its shaft to the shaft of a commercial scroll compressor in "ECO CUTE" CO2 heat pump water heater of our company. The structure of the compressor combined with expander is shown in Figure 1. The scroll compressor is located at the top, while the two-stage rotary expander is located under the compressor and the motor.

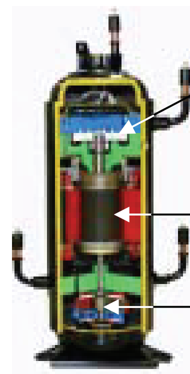

Figure 1: Compressor combined with expander

2.2 Structure and Operation of a Two-stage Rotary Expander

The two-stage rotary expander consists of two stacked rotary type fluid machines and an intermediate plate separating the first and the second stage with a connecting hole. Each stage has an independent cylinder, piston, vane, and crankpin that is a part of the crankshaft and engages with the piston.

The operation of the two-stage rotary expander is shown in Figure 2. It is designed to provide an expansion chamber surrounded by the piston and the cylinder in the second stage is larger than that in the first stage. In Figure 2 (A) through (D), the first stage is shown on the left, while the second stage is on the right, and the shaft rotation is in a clockwise direction. The two-stage rotary expander operation is shown in Figures (A) through (D), each with a 90° shaft rotation in the clockwise direction. Figure 2 (A) shows the pistons of the first and the second stage at each of the top dead center. The expansion of CO2 refrigerant starts when the shaft rotation angle θ reaches the 'suction process end angle' ($\theta s1$). At the angle, the first stage operation chamber has shut off the suction hole (Between (A) and (B)). CO2 refrigerant expands as the volume of the expansion chamber formed by the first stage operation chamber, connecting hole, and second stage operation chamber increases with the shaft rotation. The expansion process continues until the shaft rotation angle θ reaches the 'discharge process starting angle' ($\theta e2$) (Between (D) and (A)) and the refrigerant discharge starts from the discharge hole.

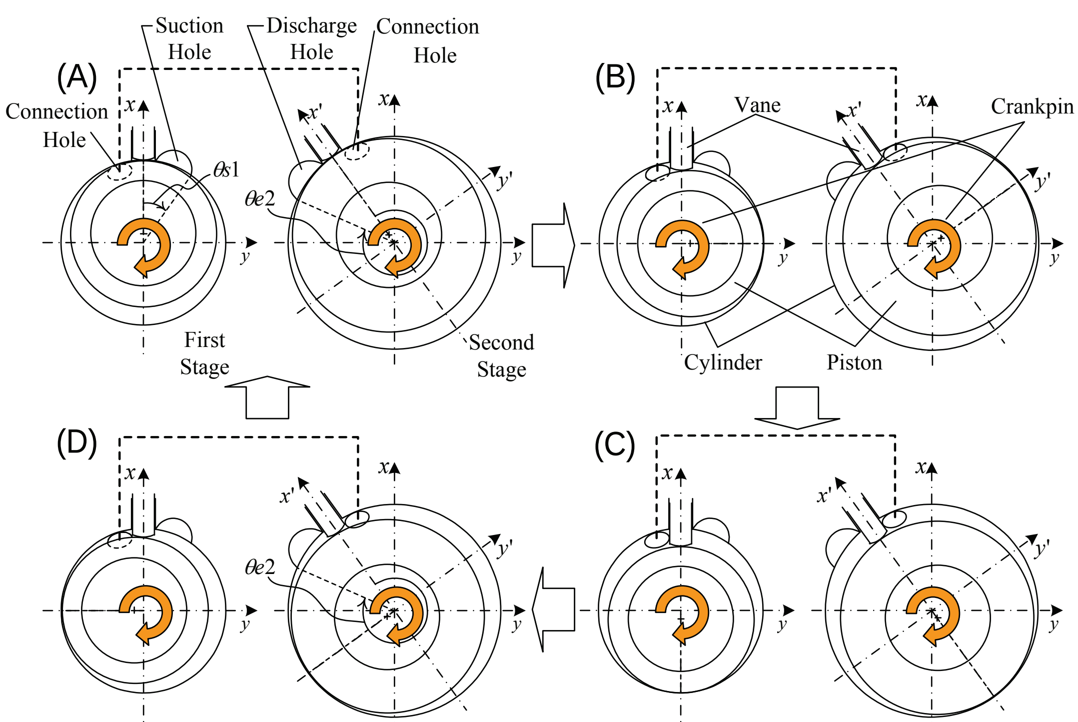

Figure 2: Operation of a two-stage rotary expander

3. ANALYSIS METHOD

3.1 Analysis Outline

By combining the dynamic mechanical analysis and the refrigerant pressure analysis of the expansion chamber with a leak analysis, the friction loss of each component of the expander, volume efficiency, and mechanical efficiency, and expander efficiency which indicate the expander performance can be calculated.

3.2 Method of Analyzing Refrigerant Pressure in the Expansion Chamber

The accurate calculation of refrigerant pressure in the expansion chamber is important for carrying out the dynamic mechanical analysis to be discussed below. Two problems exist when calculating refrigerant pressure in the expansion process. One is the development of a technique for analyzing the expansion process of CO2 refrigerant, which drastically changes itself from supercritical single phase to gas-liquid two-phase. Another one is the development of a technique for analyzing refrigerant leakage from the suction hole to the expansion chamber, and from the expansion chamber to the discharge hole.

In the conventional analysis of pressure changes in the compression process of a compressor, the refrigerant is assumed to be the ideal gas and the constant heat capacity ratio κ is applied to the formula $PV^{\kappa} = const$ (P: pressure, V: volume) for calculation purposes. However, in the expansion process of CO2 refrigerant, which changes from supercritical single phase to gas-liquid two-phase, the heat capacity ratio κ in the supercritical single phase cannot be assumed as constant because the value of κ substantially changes with a small change of pressure, and the heat capacity ratio κ is indefinite in the gas-liquid dual phase. So the formula above is not usable in the expansion process. For this reason, the refrigerant pressure analysis developed this time uses the reference database of CO2 refrigerant. That is, for every analysis time step, it gets properties of CO2 refrigerant from the standard reference database (REFPROP) of NIST and updates them in the expansion chamber.

On the other hand, the flow rate of CO2 refrigerant, which leaks from each clearance due to the pressure difference between the suction hole and expansion chamber, or between the expansion chamber and discharge hole, is calculated at every analysis time step by using the experimental formula[1] of pipe friction coefficient, which regards CO2 refrigerant as a non-compressive fluid. In addition, the leakage flow rate calculated at a specific analysis time step (shaft rotation angle θ) is assumed to affect the change of the properties of CO2 refrigerant in the expansion chamber in the next analysis step (shaft rotation angle $\theta + d\theta$).

In the refrigerant pressure analysis the calculation process is divided into two steps.

[Step 1]

Only the leakage effect is examined without considering the volume increase of the expansion chamber. In other words, the refrigerant, which leaks into the expansion chamber, is assumed to mix into the refrigerant in the expansion chamber with the specific internal energy $u_{in}(\theta)$ being kept constant during the leak-in process. The refrigerant density $\rho'(\theta)$ and specific internal energy $u'(\theta)$ after the mixture at the shaft rotation angle θ are calculated from Equation (1) and Equation (2).

$$\rho'(\theta) = \frac{M(\theta) + M_{in}(\theta) - M_{out}(\theta)}{VOL(\theta)} \tag{1}$$

$$u'(\theta) = \frac{M(\theta) \times u(\theta) + M_{in}(\theta) \times u_{in}(\theta) - M_{out}(\theta) \times u(\theta)}{M(\theta) + M_{in}(\theta) - M_{out}(\theta)} \tag{2}$$

In the above equations, $VOL(\theta)$ is the volume of the expansion chamber, $u(\theta)$ is the specific internal energy of refrigerant, and $u_{in}(\theta)$ is the specific internal energy of the refrigerant that leaks into the expansion chamber. Further, $M_{in}(\theta)$ is the refrigerant mass that leaks into the expansion chamber, $M_{out}(\theta)$ is the refrigerant mass that leaks out of the expansion chamber. $M_{in}(\theta)$ and $M_{out}(\theta)$ are calculated by using the experimental formula [1] of the pipe friction coefficient, assuming the CO2 refrigerant that leaks from the clearance to be non-compressive fluid. The refrigerant mass $M(\theta)$ in the expansion chamber is calculated by following Equation (3) based on the refrigerant mass Ms in the expansion chamber at the end of the suction process.

$$M(\theta) = Ms + \int_{\theta s1}^{\theta} M_{in}(\theta) d\theta - \int_{\theta s1}^{\theta} M_{out}(\theta) d\theta \tag{3}$$

When the values $\rho'(\theta)$ and $u'(\theta)$ are inputted to the reference database, the specific entropy $s'(\theta)$, which the influence of leakage is considered, is obtained.

[Step 2]

CO2 refrigerant in the expansion chamber assumes to expand its volume a little by tiny shaft rotational angle $d\theta$ under iso-entropy expansion process. That is, the specific entropy $s'(\theta)$ assumes to be maintained in this expansion process. When two properties of CO2 refrigerant at rotation angle $\theta + d\theta$, the refrigerant density $\rho(\theta + d\theta)$ as expressed in Equation (4) and the specific entropy $s(\theta + d\theta) = s'(\theta)$ are inputted to the reference database, the pressure $P(\theta + d\theta)$ and other properties (specific enthalpy $h(\theta + d\theta)$, specific internal energy $u(\theta + d\theta)$, etc.) can be obtained.

$$\rho(\theta + d\theta) = \frac{M(\theta) + M_{in}(\theta) - M_{out}(\theta)}{VOL(\theta + d\theta)} \tag{4}$$

3.3 Dynamic Mechanical Analysis Method

The dynamic mechanical analysis solves motion equations by numerical analysis method, that is derived from the balance of forces and moments act on the shaft, pistons and vanes, by inputting the results of the refrigerant pressure analysis described in the previous sub-section. This analysis calculates an approximate solution of the shaft rotation of the scroll compressor combined with two-stage rotary expander, and then calculates inertial forces act on the pistons and vanes with the fluctuation of the shaft rotational speed. Thus the load and friction loss on each part can be calculated. When motion equations were derived, the technique used by Imaichi et al [2] was referred to. For calculating the frictional forces act on surfaces between the vane and piston, between the piston and crankpin and of main bearing, Coulomb's friction law is assumed to be applied. The motion equation of the shaft rotation of the scroll compressor combined with two-stage rotary expander is shown in Equation (5).

$$\begin{aligned}
&\left(I_{comp} + I_s + \sum_{j=1}^{2} m_{pj} e_j^{\,2}\right)\ddot{\theta} \\
&= T_{comp}(\theta, \dot{\theta}) \\
&\quad + \sum_{j=1}^{2}\left[e_j \left\{ F_{pj} \sin\left(\frac{\theta + \xi_j}{2}\right) - F_{aj} - F_{ctj} + F_{vnj} \sin(\theta + \xi_j) - F_{vtj} \cos(\theta + \xi_j) \right\} - R_{pej} F_{etj} \right] \\
&\quad - R_s F_{bt}
\end{aligned} \tag{5}$$

The symbols used in Equation (5) are as follows. The suffix j refers to stage number of expander (j = 1, 2)

e_j	:Eccentricity	I_{comp}	:Inertial moment of the scroll compressor
F_{aj}	:Frictional force of the oil film on the top and bottom surfaces of the piston	I_s	:Inertial moment of the shaft
F_{bt}	:Frictional force of the main bearing	m_{pj}	:Mass of the piston
F_{ctj}	: Frictional force of the oil film between the piston and cylinder	R_{pej}	:Radius of the crankpin
F_{etj}	:Frictional force between the crankpin and piston	R_s	:Radius of the main bearing
F_{pj}	:Pressure differential force of CO2 refrigerant	T_{comp}	:All torques of the scroll compressor
F_{vnj}	:Frictional force between the vane and piston	ξ_j	:Displacement angle at the vane-piston contact point

3.4 Consistency with the Experiment

In order to determine the coefficient of friction of each rubbing part used in dynamic mechanical analysis, a two-stage rotary expander was manufactured (0th prototype) by setting design parameters of the expander, such as the cylinder radius, piston radius, cylinder height, etc. with no reliable reason, and tested its performance and measured the pressure in the expansion chamber (PV measurement experiment).

While referring to the study on a rotary compressor(3), the coefficients of friction were determined for ensuring that the error between friction loss obtained by PV measurement experiment and that by the dynamic mechanical analysis is 1% or less of the ideal collectable power (collectable power at 100% expander efficiency). The amount of the error is enough small compared to that of the measuring instruments used for the PV measurement experiment.

The comparisons of the analysis result and PV measurement result are shown in Figure 3 and Figure 4. Figure 3 shows the pressure of CO2 refrigerant in the expansion chamber with the volume (suction volume of expansion chamber normalized as 1) of expansion chamber on the horizontal axis, and the pressure (suction pressure normalized as 1) in the expansion chamber on the vertical axis. This figure indicates that this analysis can calculate the change in CO2 refrigerant state from supercritical single phase to gas-liquid two-phase and also indicates a good correspondence between the analysis result and the PV measurement result. Figure 4 shows the rotation speed variation of the two-stage rotary expander with time on the horizontal axis and rotation speed (average speed normalized as 1) on the vertical axis. Again, the figure indicates a good correspondence between the dynamic mechanical analysis result and the experimental result. Therefore, the analysis method is deemed sufficiently appropriate.

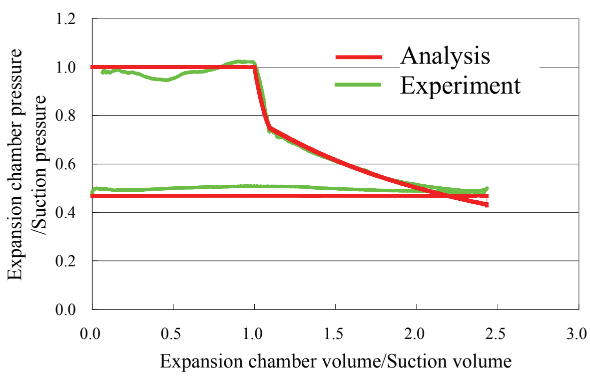

Figure 3: Pressure in the expansion chamber

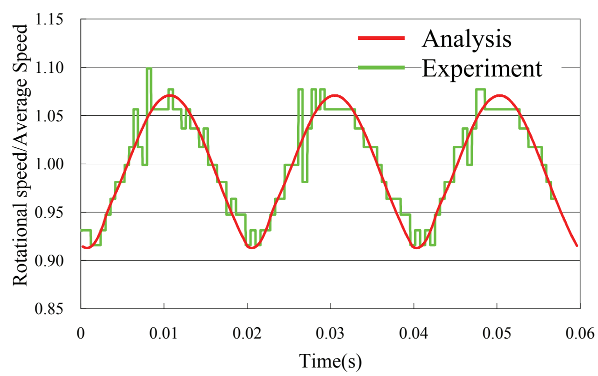

Figure 4：Rotation speed variation of the expander

4. EXAMINATION OF THE OPTIMUM DESIGN PARAMETERS

In order to maximize the expander efficiency as defined in Equation (6) by reducing friction losses of the two-stage rotary expander, the expander design parameters were optimized.

$$\eta_{ex} = \left(W_{ideal} - \sum W_{loss}\right) / W_{ideal} \tag{6}$$

In Equation (6), η_{ex} is the expander efficiency, W_{ideal} is ideal collectable power and W_{loss} is loss in each part.

Losses generated during the operation of an expander include the suction loss, phase change loss, heat receiving loss, etc. in addition to the friction loss. However, to decrease these losses except the friction loss, it is more preferable to take measures for them respectively than to optimize a set of design parameters of two-stage rotary expander. For example, to reduce the suction loss, the suction passage should be optimized. On the other hand, it is possible to reduce the friction loss by changing the expander design parameters. For this reason, we have optimized design parameters of two-stage rotary expander for the purpose of primarily reducing friction losses.

In the compressor combined with expander, the compressor and expander rotate at the same rotation speed. When the suction volume of the compressor is set, the suction volume of the expander is automatically decided according to the specific cycle condition. In consideration of the fact that the suction volume of our "ECO CUTE"'s compressor is 4.0 cc, the suction volume of the two-stage rotary expander has been decided to maximize efficiency in the winter condition when the required suction volume is smallest.

The design of experiments with an orthogonal array was applied to find the optimal design parameter of the expander. This method enables the examination of the "sensitivity" of each design parameter to the objective variable by eliminating the effects of other parameters. In this development, the developed simulation was used for experiments and the expander efficiency was selected as the objective variable.

The eight design parameters to be examined this time were selected as shown in Table 1 (The values of each level are shown by normalizing the second level as 1.0). The sensitivity of each design parameter to the expander efficiency is shown in Figure 5. From this figure, it is clarified that the following design guidelines are effective for improving the performance of the two-stage rotary expander after appropriately determining "A" (Expansion ratio).

(1) Increase B (Eccentricity of the 1st stage)
(2) Reduce E (Cylinder height of the 2nd stage)

Table 1: Design parameters of two-stage rotary expander

		1st Level	2nd Level	3rd Level
A	Expansion ratio	0.7	1.0	―
B	Eccentricity of the 1st stage	0.8	1.0	1.2
C	Cylinder height of the 1st stage	0.9	1.0	1.1
D	Eccentricity of the 2nd stage	0.8	1.0	1.2
E	Cylinder height of the 2nd stage	0.7	1.0	1.3
F	Radius of crankpin of the 1st stage	0.9	1.0	1.1
G	Radius of crankpin of the 2nd stage	0.9	1.0	1.1
H	Radius of the main bearing	0.9	1.0	1.1

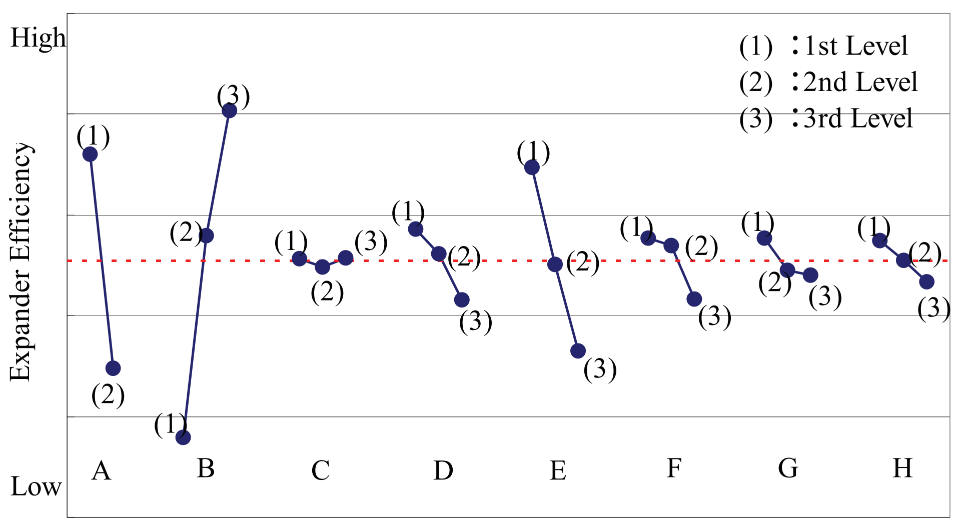

Figure 5: Sensitivity of each design parameter to expansion efficiency

5. PERFORMANCE VERIFICATION BY EXPERIMENTS

A two-stage rotary expander has been designed and manufactured based on the design guidelines described in the previous section. With the first prototype, a total of four types of expanders were designed in consideration of restrictions of manufacturing to verify the validity of the design guidelines by varying the values of design parameters (B) and (E), which have been assessed as having a higher sensitivity. The value of the design parameter (B) was taken 1.0 and 1.2, and the value of (E) was taken 0.7 and 1.0 (these values are normalized by the value of the second level in Table 1). The sensitivity of design parameters (B) and (E) to the expander efficiency obtained from the performance examinations of the four types of expander is shown in Figure 6. Figure 6 also shows the

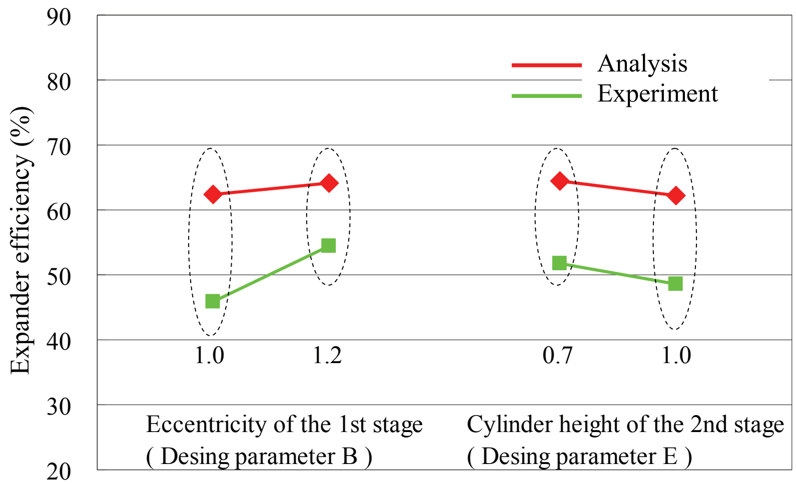

Figure 6: Sensitivity of design parameters B and E to expander efficiency

sensitivity of design parameters (B) and (E) to the expander efficiency obtained by this analysis. In Figure 6, the sensitivity of (B) and (E) to the expander efficiency indicates the same tendency between the analysis and experiment, but there's a substantial difference in the value. The cause of the difference in the value of the expander efficiency between the analysis and experiment has been investigated in detail. First, the rubbing traces on the piston of the second stage and the intermediate plate separating the first and the second stage were identified. It showed that a large friction loss was generated between the piston and intermediate plate that had not been assumed in the 0th prototype, which had been used to adjust the coefficients of friction. Second, the shape of the suction hole was changed in the first prototype from that of the 0th prototype, and as a result, the suction loss was found to be substantially large.

Therefore, in the second prototype, to set the optimal clearances between the pistons and intermediate plate by taking the deformation of the intermediate plate into consideration, the deformation of the intermediate plate was analyzed and quantified by using the commercial structural analysis software ‘I-deas’. An analysis example of the intermediate plate is shown in Figure 7. The figure shows a deformation of the intermediate plate toward the second stage side due to the pressure difference between the first and second stage. The suction hole shape was also modified to reduce the suction loss. The performance of the second prototype after implementing the improvements achieved the initially targeted expander efficiency of 60% (Table 2). Moreover, we confirmed the improvement of the cycle COP by 6% by integrating the developed expander into the cycle.

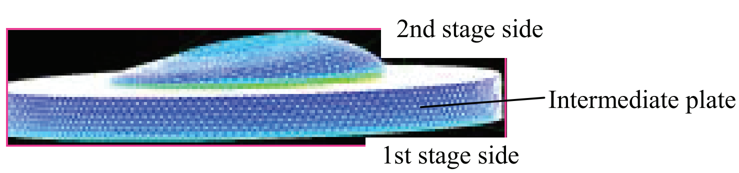

Figure 7: An example of deformation analysis result of the intermediate plate by using ‘I-deas’

Table 2: Performance evaluation results

	Analysis	Experiment
Expander efficiency (%)	58	60

6. CONCLUSIONS

In order to develop a two-stage rotary expander efficiently, the performance analysis simulation has been developed by combining the refrigerant pressure analysis and the dynamic mechanical analysis. Then applying the design of experiments based on an orthogonal array, the "sensitivity" of the expander design parameters to the expander efficiency was examined by using this simulation for extracting the optimum set of design parameters. The described process produced the following results.

(1) The refrigerant pressure analysis carried out by obtaining and updating properties of CO2 refrigerant from the standard reference database (REFPROP) of NIST, and calculating the leakage flow rate of CO2 refrigerant in each clearance by using the experimental formula(1) of pipe friction coefficient that regards CO2 refrigerant non-compressive fluid at every analysis time step, enables the analysis of the phase change process of CO2 refrigerant from supercritical single phase to gas-liquid two-phase and is deemed appropriate by comparing with the PV experiment results.

(2) The rotational motion equation of the shaft of the scroll compressor combined with two-stage rotary expander developed this time has been deemed reliable by comparing the values with the shaft rotation speed variations obtained from the PV measurement experiment. Further, the dynamic mechanical analysis to calculate the loads act on the pistons, vanes and shaft by taking the varying inertial force of the pistons and vanes with the shaft rotation into consideration, and then calculate the friction loss in each rubbing part, has been made reliable by adjusting the coefficients of friction to fit the analysis result to the total friction loss obtained by the PV measurement experience.

(3) The design parameters for the input data of the performance analysis simulation were changed based on the orthogonal array, and the "sensitivity" of each design parameter to the expander efficiency was examined by this simulation and the optimum set of design parameters was extracted. As a result, an expander efficiency of 60% has been achieved after two prototyping cycles in substance. The heat pump cycle integrated with the developed expander has been verified to improve COP by 6%.

REFERENCES

1. Oku, T., et al, 2005, Basic study of gas leakage flow through small clearances in scroll compressors, *Proceedings of the 2005 JSRAE Annual Conference*, C312
2. Imaichi, K., et al, 1982, Vibration analysis of rotary compressors, *Proceedings of Purdue Compressor Technology Conference*, P.275-282
3. Shintaku, H., et al, 2000, Characteristics evaluation of new type rotary compressor, *Proceedings of the 34th the federation conference of air-conditioning and refrigerating*, p.57-60, 2000