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1. ABSTRACT

This paper presents the development of the high-efficiency and low-vibration rotary compressor for HFC-410A air-conditioners.

As the pressure of HFC-410A is 1.6 times higher than that of HCFC-22, the rotary compressor in the conventional design has the lower efficiency by the larger leakage from the compression chamber to the suction chamber. The length and the diameter of the cylinder have been optimized to reduce the leakage and to have the high reliability at the sliding surfaces between the blade and the cylinder.

The rotational vibration has been reduced by the improvement of the control method, which generates the motor torque adjusted to the fluctuation of the gas compression torque.

2. INTRODUCTION

HCFC-22 is changing to HFC-410A in the room air conditioner. Especially the high-class air conditioner using a scroll compressor or a two-cylinder rotary compressor has changed to HFC-410A in 20th century.

Itoh et al.[1] and Mitarai et al.[2] reported the performance of HFC-410A scroll compressor. Hayano et al.[3], Higuchi et al.[4], Fukuda[5] and Ishii et al.[6] also reported the performance of the HFC-410A two-cylinder rotary compressor. However, in the low-cost single-cylinder rotary compressor which occupies a big market share, it is hard to say that the alternation from HCFC-22 is progressing enough. Alternation to HFC-410A of air conditioners that use the single-cylinder rotary compressor is one of the most important subjects in the viewpoint of the earth environment. The development of the single-cylinder rotary compressor with high efficiency and low cost was expected as solution of the above-mentioned subject. However, since a single-cylinder rotary compressor has the large vibration by torque fluctuation, the minimum rotating speed becomes higher than a two-cylinder rotary compressor.

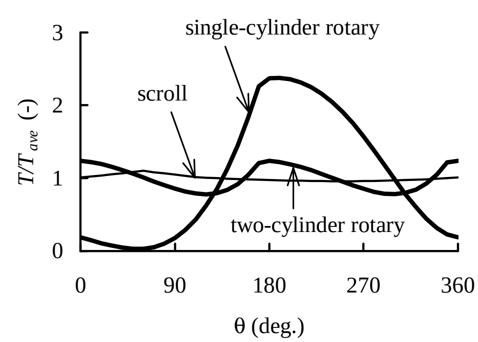


Fig. 1 shows the torque fluctuation of the scroll compressor, two-cylinder rotary compressor, and the single-cylinder rotary compressor in high-pressure

P_h=2.25MPa and low-pressure P_l=1.16MPa. The X-axis is the rotating angle θ and the Y-axis is the non-dimensional torque T/T_{ave}. T_{ave} is defined in the average torque. The torque fluctuation of the scroll compressor is less than 15%, and that of the two-cylinder rotary compressor is less than 30%. On the other hand, the torque fluctuation of the single-cylinder rotary compressor is more than 100%. Therefore, in the conventional single-cylinder rotary compressor, the compressor drive range is narrower than the capacity range required from an air-conditioner unit, and the efficiency of the air-conditioner with the single-cylinder rotary compressor becomes lower than that of one with the two-cylinder rotary compressor.

From this background, this paper reports mainly the motor technology, which aims at the drive range expansion to the low speed of the single-cylinder rotary compressor.

3. STRUCTURE OF HFC-410A SINGLE-CYLINDER ROTARY COMPRESSOR

3.1 Cylinder Dimension and Performance Characteristic

Reports of Hayano et al.[3], Higuchi et al.[4], Fukuda[5] and Ishii et al.[6] showed the characteristics of the HFC-410A two-cylinder rotary compressor efficiency examined with the parameter such as the bore diameter *D* and height *L* of the cylinder. However, there is no example of the area divided by limitation lines in which durability, assembly and size are realizable in *D*-*L* plane.

Furthermore, there is almost no compared example about the efficiency in the same suction volume of the single-cylinder and the two-cylinder rotary compressor. The examination result of the efficiency of the rotary compressor with the capacity of 2.2kW here is shown.

The suction volume V_{th} of the rotary compressor is given by the following equation.

$$V_{th} = \pi L\,(R_c^2 - R_o^2) \qquad \ldots(1)$$

R_c: cylinder radius $=D/2$, R_o : rotor radius

The eccentricity ε is given by the following equation.

$$\varepsilon = R_c - R_o \qquad \ldots(2)$$

The following limitations exist to determine the dimensions of the rotary compressor.

(i) Diameter limitation: $R_c+L_b+R_s+B=d/2$ …(3)

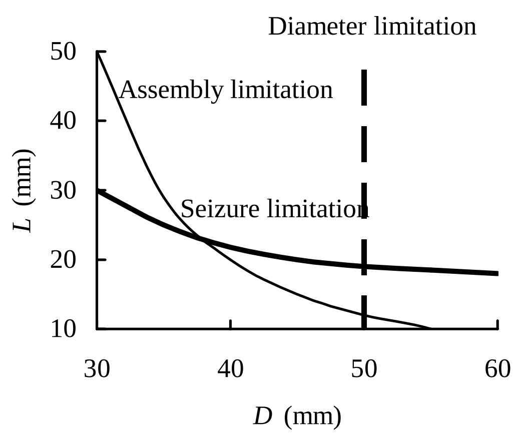
d : compressor diameter, R_c : cylinder radius, L_b : blade length,
R_s : broach stop hole radius, B: bridge section length

(ii) The seizure limitation of the blade side (*PV* limit): $P_b\,V_b(\theta) < Q_b$ …(4)

P_b : load, $V_b(\theta)$: velocity in rotation angle θ, Q_b : seizure limitation

(iii) Assembly limitation: $R_j+\varepsilon < R_p$ …(5)

R_j : journal-bearing radius, ε : eccentricity,
R_p : pin radius



The area, which fulfills conditions of (i)-(iii), is shown in fig. 2. *X*-axis is the cylinder bore diameter *D* and *Y*-axis is the cylinder height *L*. Journal bearings are designed in fluid film lubrication, even if any *D* and *L* is chosen. it is possible to make film thickness into the same value by adjusting the bearing gap. For this reason, the rotary compressor with equal *PV* value of the blade side can be considered to the compressor with the same durability.

3.2 Mechanical Loss

Journal-bearing loss and loss of the blade side occupy most mechanical losses. Journal loss W_j is given by the following equation.

$$W_j=\mu_j F_j V_j \qquad \ldots(6)$$

μ_j : coefficient of friction(=0.004), F_j : journal load, V_j : journal velocity

Loss W_b of the blade side is given by the following equation.

$$W_b=\mu_b F_b V_b \qquad \ldots(7)$$

μ_b : coefficient of friction(=0.15), F_b : blade side load, V_b : blade velocity

3.3 Indicative Loss

Indicative efficiency can be divided roughly into leakage losses, suction pressure drop loss, discharge pressure drop loss and re-compression loss. Except leakage losses, it can be made almost equal by making the same diameter of the inhalation pipe, the diameter of the discharge pipe, and dimensions of the discharge valve. Therefore, leakage losses are most strongly influenced of cylinder dimensions. The leakage paths are as follows.

(a) The gap between the cylinder side and the rotor side

(b) The gap between the differences of rotor height and cylinder height

(c) The gap between the blade slot and the blade thickness

The lubricating oil leaks through the leakage path (b) and (c). The refrigerant which dissolved into the lubricating oil makes the leakage loss. This loss W_L is proportional to the product of cylinder height L and clearance δ.

$$W_L=C\,L\,\delta \qquad \ldots(8)$$

C : proportionality constant, δ: clearance

When δ is set constant, indicative loss is proportional to the cylinder height L. Fig. 3 shows the calculation result of the compressor efficiency, which is given by the product of the indicative efficiency, mechanical efficiency and motor efficiency (=0.92 fixed value was assumed) in the ASHRAE-T condition.

The compressor efficiency η becomes high when cylinder height is low and the compressor efficiency becomes high when the diameter of the cylinder is large. In order to verify this property, two kinds of compressors, which had different cylinder height, were examined (test A and test B). Dimensions of compressors are shown in table 1. Fig. 4 shows experimental results of the efficiency ratio η_A/η_B in the ASHRAE-T conditions. There is 1-3% difference of efficiency between the two compressors according to the difference of cylinder height. From this result, prediction of fig. 3 was verifiable with the experiment.

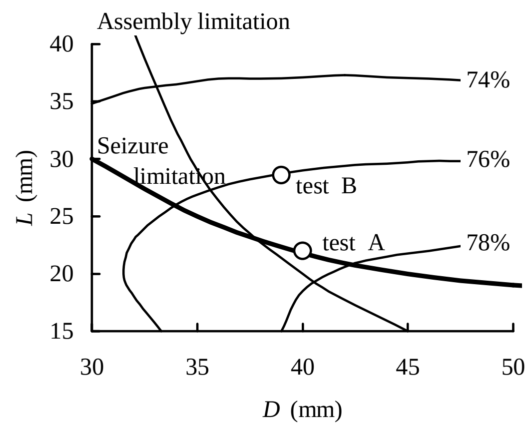
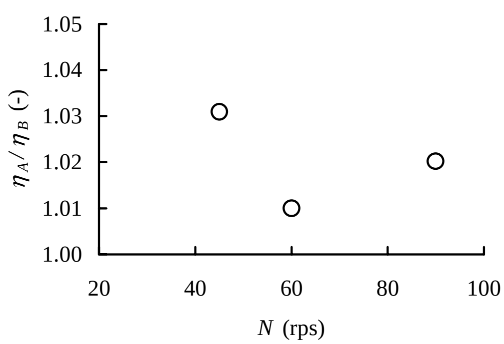


Table 1 Compressor dimensions

	test A	test B
D (mm)	22	28
L (mm)	40	39



3.4 Efficiency Comparison of Single-Cylinder Rotary and Two-Cylinder Rotary

The two-cylinder rotary compressor is in the tendency for total cylinder height to become higher than the single-cylinder rotary, because the suction pipe area of the two-cylinder sum total is the same as that of the single-cylinder rotary compressor.

The following equation is established between suction pipe diameter *Ds* of the single-cylinder and two-cylinder suction pipe diameter Ds_2.

$$2\pi Ds_2^{\,2}/4 = \pi Ds^2/4 \qquad \ldots(9)$$

$$Ds_2/Ds=(1/2)^{0.5}=0.71 \qquad \ldots(10)$$

In this case, the sum of two-cylinder suction pipe diameter serves as 1.4*Ds*. From fig. 3, compressor efficiency become lower as cylinder height becomes higher. That is why the efficiency of the two-cylinder rotary compressor will be inferior to the efficiency of the single-cylinder rotary compressor.

However, the two-cylinder rotary compressor has the merit with the small torque fluctuation, as shown in fig. 1. Conversely, if the vibration at the low speed is reduced, the single-cylinder rotary compressor excels the two-cylinder rotary in all fields.

4. VIBRATION REDUCTION TECHNIQUE OF SINGLE-CYLINDER ROTARY COMPRESSOR

The technique of controlling the output torque of the motor existed with birth of the motor. The report of Inoue et al.[7] is the example which applied the torque control of the motor to positioning control. The vibration reduction technique of the single-cylinder rotary compressor which is driven with the brush-less DC motor is shown analytically and experimentally in detail by Nakamura et al. [8,9].

Nakamura et al.[8,9] has proposed the real-time system which feeds back the amount of torque corrections by the rotating speed fluctuation or acceleration fluctuation in that paper. The rotating speed fluctuation of the compressor or the acceleration fluctuation is detected from the induced voltage of the motor.

This system is the excellent one as the vibration reduction technique according to the load of the compressor. However, there are following demerits. The determination algorithm of the feedback gain, which reconciled the system stability and the vibration reduction, is complicated. High-speed CPU is necessary in order to calculate the amount of the correction for torque control.

The above-mentioned technique does not suit the purpose which offers the low-cost single-cylinder rotary compressor.

In this paper, the control technique referring to the table of the amount of the torque correction decided beforehand was developed.

4.1 Compressor Vibration Analysis

Fig. 5 shows the torque control block diagram. The control method is presented as follows.

The average rotating speed N^* of the compressor is detected from the induced voltage of the motor.

N^* is compared with N, and the voltage of correction V_{av} is decided in PI controller. On the other hand, the torque reference table is referred and the torque control voltage V_{tq} is given. The total voltage correction V_c is applied to the drive unit.

In this system, the feedback from the motor is only the average velocity. Low-speed CPU can control this system

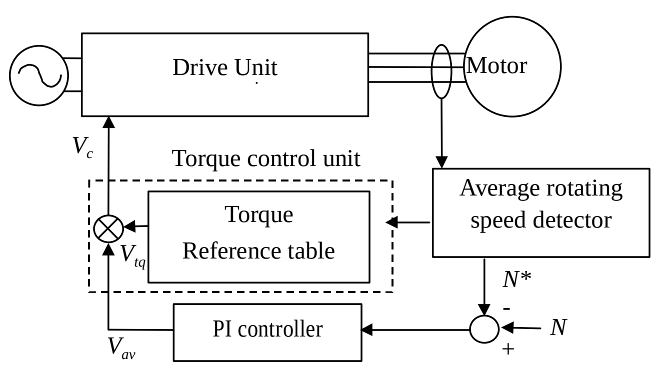


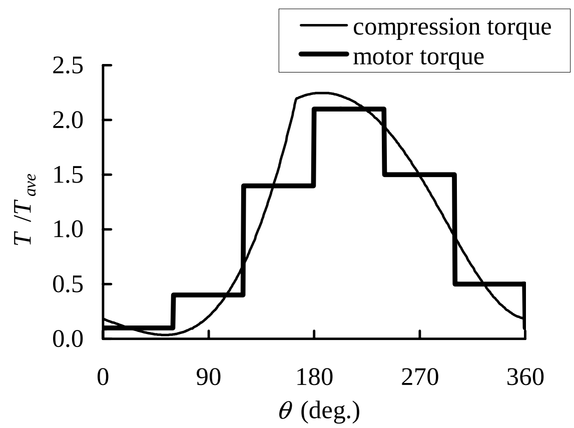
Fig. 5 Block diagram of motor drive unit with torque control.

since there is little operation. Moreover, since the drive of the brush-less DC motor is usually equipped with rotational speed detection devices of the motor, additional devices to the circuit are unnecessary. The effect and stability of control system depends on the amount of corrections of the referring table. In order to determine this amount of corrections, the vibration analysis of the compressor was estimated. The vibration of the shell centering on the compressor center axis is expressed with the following motion equation.

$$I\, d^2\theta / dt^2 + c\, d\theta/dt + k\,\theta = \Delta T = T_m - T_c \qquad \ldots(11)$$

I is the moment of inertia of the fixed system ; =278x10^{-4} kg-m^2 , k is spring constant ; =298.8 Nm/rad , c is damping-coefficient ; =2.8 Nms.Constant values of I, k and c is experimental values.T_m is the motor output torque and T_c is the compression torque. ΔT is defined in T_m-T_c.

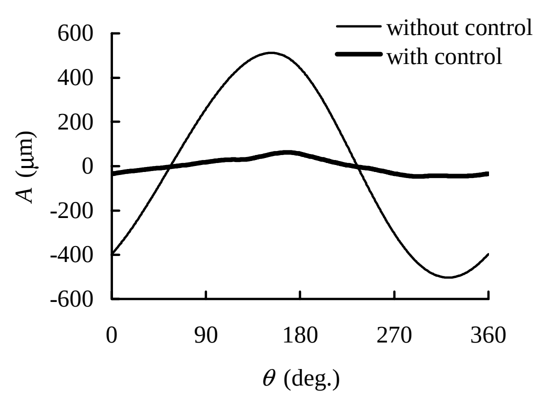
This motion equation was solved in periodic boundary conditions until obtaining the quasi-static solution. The conversion to the amplitude of the suction pipe was given by the following formula.



4.2 Optimal Torque Pattern of Vibration Reduction

Fig. 6 shows the compression torque of the single-cylinder rotary compressor and the motor output torque with the torque control. One rotation [360 degrees] were equally divided into six, and the motor output torque was changed in each section. The motor is the DC blush-less motor with the four poles, as for this reason, one rotation of the machine is electrically equivalent to two rotations. If this is written typically, each phase of 3 phase power sources will be made into U phase, V phase, and W phase.

In order for the mechanical one rotation, the current flows U1-V1-W1-U2-V2-W2 and the magnetic field rotation is required. In the brush-less DC motor, the motor output torque is proportional to the current. The motor output torque pattern was decided in agreement with the section average of the compression torque.



The motor output torque T_{mi} in Section i (i=1, 6) is given by the following equation.

$$T_{mi} = \frac{1}{\theta_{i+1} - \theta_i} \int_{\theta_i}^{\theta_{i+1}} T_C\, d\theta \qquad \ldots(12)$$

The compression torque of the rotary compressor changes by the suction pressure and the discharge pressure in the same rotating speed. However, in the control technique of referring to the table, torque is not corrected by change of pressure. That is why the correction table was created on the conditions that the fluctuation of compression torque is the largest.(High-pressure P_h = 2.25MPa, low-pressure P_l = 1.16MPa) This technique is hereafter called the section average torque method.

The suction pipe amplitude A in the case of compressor rotating speeds N=20rps, high-pressure P_h= 2.25MPa, and low-pressure P_l = 1.16MPa is shown in fig. 7.

The result which carried out the fast Fourier transform (FFT transformation) of the torque difference by the section average torque method is shown in fig. 8. Although the amplitude of the torque fluctuation is reduced only up to the about 0.74 times by the section average torque method, the amplitude of the suction pipe is reduced by the 0.11 times. The reason why the amplitude of the suction pipe is reduced can be explained from fig. 8.

The fluctuation degree n of the torque difference equal to 1st without the control of the motor output torque. On the other hand, the fluctuation degree n of the torque difference has the 5th and 7th mainly with the control of the motor output torque. Since the amplitude reduces in proportion to $1/n^2$, the vibration decreases by the torque control.

The control start angle is another variable which determines the torque pattern of the section average torque method. Fig. 9 shows the analytical result of the suction pipe amplitude in compressor rotating-speed N=20rps, high-pressure P_h = 2.25MPa, and low-pressure P_l = 1.16MPa. In Θ- Φ = 35 deg, the amplitude becomes the smallest. Θ : mechanical rotation angle, Φ : electric rotation angle

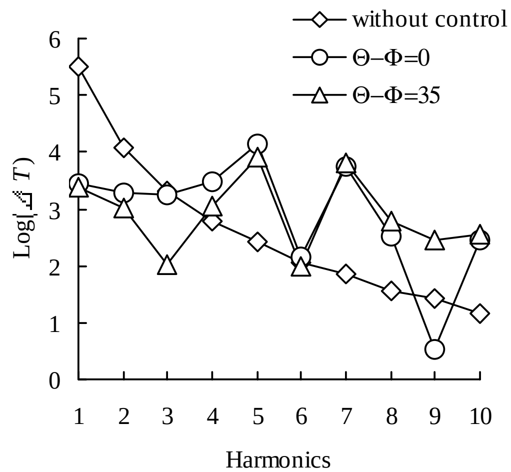


Fig. 8 Torque harmonics.

4.3 Loss Evaluation

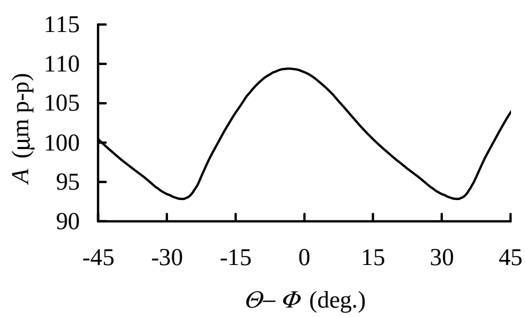
The torque control pattern is set up by the compromise point between the motor efficiency decline increasing the current and the effect of vibration reduction. Motor efficiency was estimated by the following formula. The motor output torque T_m is proportional to motor current I_m.

$$T_m \propto I_m \qquad \text{…(13)}$$

Joule loss of the motor W_m is proportional to the 2nd power of the current I_m.

$$W_m = I_m^2 R_m \qquad \text{…(14)}$$

R_m : Resistance of the motor



The vibration reduction effect and the motor loss are in inverse proportion. The torque pattern which was the minimum loss of the motor was chosen from what satisfied the tolerance amplitude of the suction pipe in the use range. For this reason, the torque pattern with the control start angle Θ-Φ =0deg was adopted as an actual control pattern.

4.4 Experiment

4.4.1 Examination equipment

The scheme of the single-cylinder rotary compressor which was examined in the vibration test is shown in fig. 10. The compressor is connected to calorimeter with the flexible pipe. Vibration of the compressor was detected by the acceleration pickup attached to the inhalation pipe. The acceleration pickup was attached in radial from the center of the compressor at 120mm. The response frequency of the acceleration pickup is 10kHz.

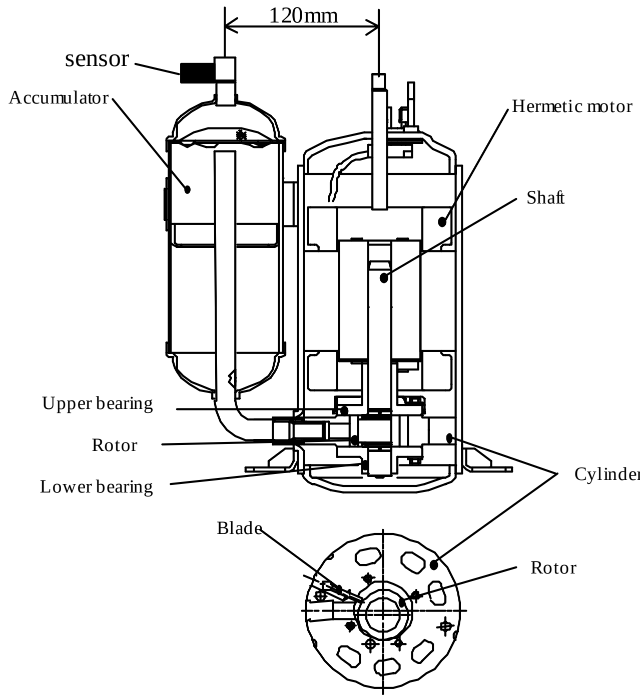
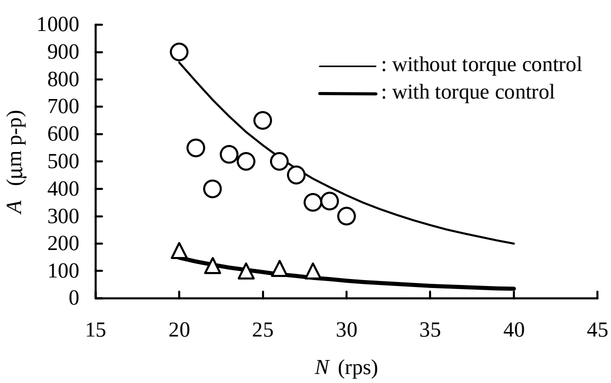


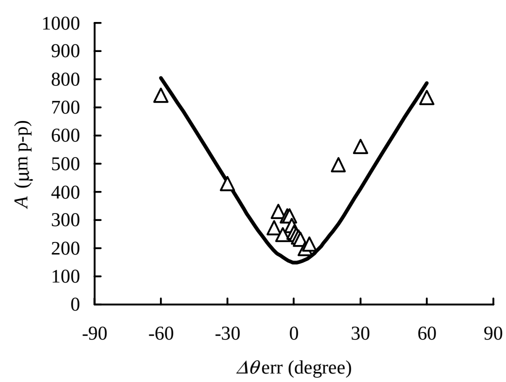
Fig. 10 Scheme of the test compressor.

4.4.2 Results and discussion

Fig. 11 shows the analysis and the experiment result of the suction pipe amplitude in P_h=2.25MPa and P_l=1.16MPa. From fig. 11, the analysis result and the experiment result have a good agreement. Furthermore with the torque control, the result confirmed that the amplitude could be reduced to one fifth. The amplitude by the assembly error was investigated.

Fig. 12 shows the vibration to assembly angle error $\Delta\Theta_{of}$ the motor stator and the compression section. Also in the assembling error, the prediction and the experiment result showed good agreement. The experiment result provided the evidence that the increase of the suction pipe vibration at ±5 degrees of assembling tolerance zones was not problem. Moreover, the control system which reads the motor output torque pattern in the reference table is superior to the feedback control system in the vibration reduction of the single-cylinder rotary compressor cheaply.





5. CONCLUSIONS

The efficiency map which made the cylinder bore and cylinder height the parameter was shown. The dimension which becomes the highest efficiency in the rotary compressor was shown. The efficiency difference of the single-cylinder and the two-cylinder rotary was shown, and the efficiency of the single-cylinder rotary was superior to the efficiency of the two-cylinder rotary compressor was shown.

The motor output torque control method using the reference table was developed for vibration reduction at the low speed which is the demerit of the single-cylinder rotary. The torque pattern which suited the rotary compressor was shown. It was shown that evaluation of not only oscillating reduction but also motor loss is required for the determination of the torque control pattern. The results of analysis and experiment had good agreement and it was confirmed that the vibration could be reduced to one fifth. It was estimated by analysis and experiment that the vibration increased by the assembling error of the motor. It became clear that assembling by ±5deg is required.

The low-cost and high-efficiency single-cylinder rotary compressor with the low vibration was developed. The conversion to HFC-410A is promoted and social responsibility of preserving the earth environment was performed.

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