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STUDY OF TWIN ROTARY COMPRESSOR FOR AIR-CONDITIONER WITH INVERTER SYSTEM

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ABSTRACT

The continuing consumer pursuit of greater comfort in the everyday life stresses the need for air conditioners that offer better-quality air conditioning with less energy bills and less noise. The essential performance of air conditioners depends on the characteristics of the compressor, the heart of any air conditioner. Particularly, inverter-driven compressors are required to provide a broad variable working rpm range and offer high efficiency, low noise, and low vibration throughout such range.

Toshiba has developed a high-performance twin rotary compressor having two compression chambers. A phase difference of 180° in their mutual compression timing improves the dynamic balancing and torque ripple characteristics of the compressor to provide silent, high-efficiency operation over a broad rpm range, from low to high.

INTRODUCTION

General family air conditioners(hereafter simply called air conditioners), among which the heater/cooler heat pump type is dominant, have come into such widespread use in the consumers'everyday lives that one air conditioner can now be found in almost every room in a house. Since Toshiba pioneered in the marketing of an inverter air conditioner in 1981 for the first time in the world, inverter air conditioners have led the subsequent evolution of general family air conditioners in pace with the growing needs for better air conditioning comfort and energy conservation.

The essential performance of air conditioners is determined by the characteristics of the compressor, the heart of any air conditioner. Two major requirements that face inverter-driven compressor are: a broad variable working rpm range, and high efficiency, low noise, and low vibration throughout such range. Previous inverter air conditioners used a single-cylinder single rotary compressor operating on a single-rotation, single-compression principle. This compressor structure, however, was short of assuring the inverter air conditioner of full performance, because it was liable to vibration, noise, and shaft whirling associated with rotational imbalances, limiting the minimum working rpm, which relates closely to energy conservation and comfort, and the maximum working rpm, which relates closely to heating/cooling startup speeds. Despite the subsequent advent of various analysis techniques and continuing pursuit of optimization designs to improve compressor characteristics, a variable working rpm range, and high-efficiency, low-noise, and low-vibration requirements have become increasingly more incompatible tasks.

To fill the need for added performance for inverter-driven compressors, Toshiba has developed a twin rotary compressor for use in general family inverter air conditioners, which is furnished with two compression chambers to produce a phase difference of 180° in their mutual compression timing. In the product design scene, the bending deformation behavior and bearing load characteristics of the rotor shaft during whirling were dynamically analyzed by directly calculating the bearing force as part of newly developed axial behavior analysis procedures with the Reynolds equations so as to improve both reliability and performance. Structural and efficiency analyses are also reflected in the compressor design to employ a new frame clamping method and a dual-suction structure for added performance.

With significantly improved rotational balancing and compressive torques, the twin rotary compressor operates from as low as 12rps(720rpm) to as high as 150rps (9000rpm). It has not only brought about a five- to ten-fold more advance in the variable working rpm range(maximum/minimum rpm), but has realized high efficiency, low noise, and low vibration throughout the working rpm range.

Precision manufacturing technologies developed in the meantime support the mass production of the twin rotary compressor in all its phases, from machining to assembly, to ensure higher product performance.

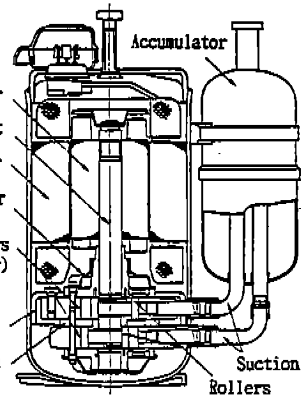
PRODUCT SUMMARY

Table 1 summarizes the specifications of the twin rotary compressor for use in inverter air conditoners in comparision with Toshiba's single rotary compressor of the comparable capacity rating. Figure 1 is a cross section of the twin rotary compressor. Principal features of the internal structure of the twin rotary compressor are described below.

Table 1

Specifications for compressor

Specification	Twin-rotary compressor	Single-rotary compressor
Cylinder diameter ×Stroke (mm)	41×3.5	41×4.0
Number of cylinders	2	1
Displacement(cc/rev)	13.3	13.1
Motor output (w)	750	750
Variable frequency range (rps)	12~150	30~135
Total hight ×Case outside diameter(mm)	272×115	272×115



Principal cross-section of twin-rotary compressor

Figure 1

Compression Chamber Structure

A partition plate separates the compression space into two chambers, upper and lower. The eccentric axes of the shaft rotated by the motor are located at 180 opposite positions in the upper and lower compression chambers. Each rotation of the shaft causes the rollers in the upper and lower compression chambers to compress and send out refrigerant gas twice with a phase difference of 180°.

Mechanical Unit Clamping Method

A new frame clamping method is used to secure the mechanical unit of the compressor to the enclosed cylindrical case. This method involves less cylinder deformation during assembly than did the previous practice of welding the cylinder to the container directly. The resultant reduced clearances between the cylinder and the roller, blade lessen the leakage of refrigerant gas into the compression space, a key to achieving enhanced rotary compressor performance. The frame geometry and the positions at which the frame and the cylindrical case are fixed have been established for optimum noise characteristics by using eigenvalue analyses allowing for the natural frequency of the mechanical unit.

Suction Pipes

As can be seen from Figure 1, suction takes place in a dual-suction structure, in which refrigerant gas is sucked into the upper and lower compression chambers through independent pipes. The supercharging effect improves volumetric efficiency in the high-speed revolution zone.

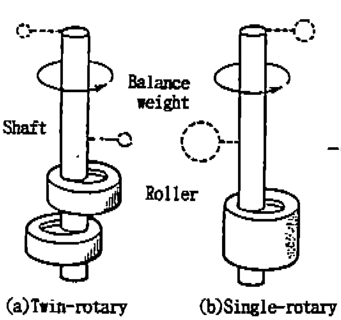
Motor

A high-efficiency motor has been developed and mass-produced to fit into the twin rotary compressor. Coupled with the developement of a high space-factor winding assembly technology for the mass production line and the use of low-loss electromagnetic steel sheets, the motor has demonstrated 2-3% higher efficiency in the entire working rpm range than the previous motors.

PERFORMANCE

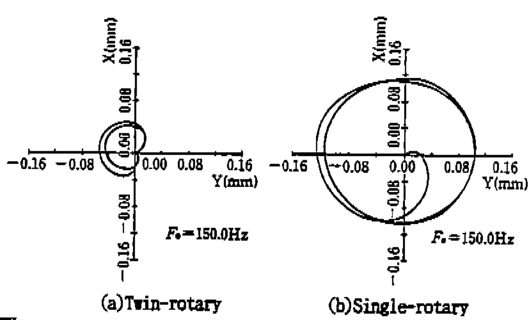
Low Vibration

The vibration of the rotary compressor in operation can be broken down into two sources: vibration in the normal direction of the cylindrical case resulting from the mass imbalance in the shaft rotor rotation system, and vibration in the rotational direction resulting from the pulsation of compressed refrigerant gas. The mass imbalance in the shaft rotor rotation system is offset by installing balancing weight in the upper and the lower part of the rotor to keep the unbalanced force of rotation and the unbalanced force of moment in balance as shown in Figure 2.



Mechanical balancing model

Figure 2



The orbits of whirling of the rotor top in a twin- and single-rotary compressor are compared.

Figure 3

The unbalanced masses include the eccentric masses of the eccentric axes and roller. The twin rotary compressor has equal unbalanced masses located at 180° opposite positions, which exert mutually centrifugal forces in mutually reverse directions to cancel the unbalanced force of rotation. Thus, the problem can be reduced to a mere installation of balancing weights to correct the unbalanced force of moment in the distance between the eccentric axes. Consequently, the twin rotary compressor is held to less than one-tenth of the mass of a single rotary compressor, thereby suppressing whirling of the rotor top during high-speed revolution and permitting high-speed operation.

Figure 3 gives a typical numeric calculation of the amount of whirling of the rotor top in a twin rotary compressor in comparison with a single rotary compressor. An about 50% cut in the amount of whirling in the twin rotary compressor below that of the single rotary compressor attests to its exceptionally improved dynamic balance.

The amplitude of the vibration in the rotational direction resulting from the pulsation of compressed gas can be expressed as a solution to an equation of motion, in which the compressor is thought of as a rigid body:

$$R(t) = Q(t) \times L \tag{1}$$

$$Q(t) = \frac{1}{I_p \omega_c} \int_0^t T(\tau)\, e^{-n^*(t-\tau)} \sin\{\omega_c (t-\tau)\}\, d\tau \tag{2}$$

where

$R(t)$: Amplitude of rotational vibration $Q(t)$: Angular displacement
L : Distance from the center of revolution of the compressor
I_p : Moment of inertia around the axis of center of revolution of the compressor
ω_c : Natural vibration in the presence of torsional damping constant C
$T(\tau)$: Torque variation. τ : variable
n^* : $\gamma \times \omega_n$, γ : Damping constant ratio
ω_n : Natural vibration in the absence of torsional damping constant C

If the torque of a single rotary compressor is $Ts(\omega t)$, a Fourier transform of the periodic function determined from the geometric shape of the compression space translates it into:

$$T_s(\omega t) = \sum_{n=1}^{\infty} T_n \cos(n\omega t + \Phi_n) \tag{3}$$

where

T_n : Torque of the n-th order. Φ_n : Phase angle of the n-th order

Similarly, the torque $Tt(\omega t)$ of twin rotary compressor can be expressed as:

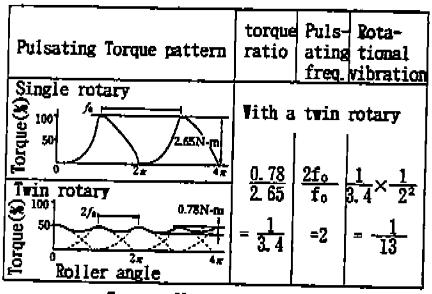
$$\begin{aligned} T_t(\omega t) &= T_s(\omega t)/2 + T_s(\omega t - \pi)/2 \\ &= \sum_{n=1}^{\infty} T_{2n} \cos(2n\omega t + \Phi_{2n}) \end{aligned} \tag{4}$$

Thus, only the even-numbered components of pulsation of compressed gas in the single rotary compressor remain. By disregarding weak torque components of the third order and higher orders, the above equation can be approximated as:

$$|T_t / T_s| = T_2 / \sqrt{T_1^2 + T_2^2} \tag{5}$$

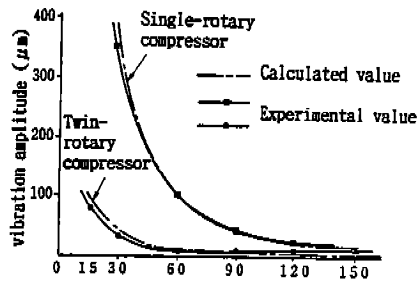
Figure 4 presents calculations of the pulsation torque pattern and the amplitude of vibration in a twin rotary compressor compared with a previous single rotary compressor. Figure 5 provides the calculated and measured values of the amplitude of vibration relative to the working rpms.

The rotational vibration in the twin rotary compressor has been cut sizably by synergetic effects of reductions in torque pulsation and the pulsating frequency when compared with the single rotary compressor.
With single rotary compressors, vibration tends to increase greatly as the compressor run at lower speed. As a result, the low-speed operation of the single rotary compressors are limited to more or less than 30 rps due to problem relating to the stress of the piping in the condensing unit and noise resulting from vibration. The twin rotary compressor now developed, on the other hand, assures sufficient bearing lubrication while running at even lower speeds, allowing operation at as low as 12 rps while suppressing vibration.



Torque fluctuation patterns and vibration amplitudes

Figure 4



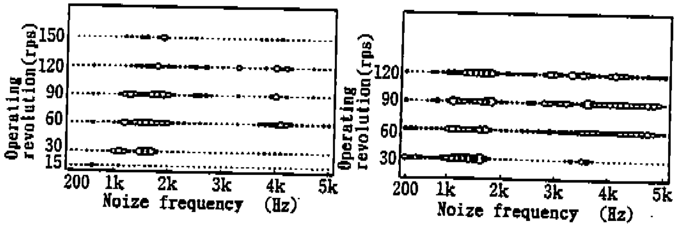
Twin- and single-rotary compressor rotational vibration amplitudes

Figure 5

Low Noise

While air conditioners are built to serve the goal of creating better living comfort, the persistent user always calls for further cuts in the operating noise of the air conditioner. The noise of the condensing unit is largely influenced by the operating noise of the compressor.

For added noise analysis accuracy, a newly developed method of evaluation was used for acoustic power level measurement in the present developement of the twin rotary compressor. Among the analysis software specifically developed are: an acoustic power spectral map, which extracts principal noises readily, providing ready insight into their characteristics; an acoustic time-series analysis, which offers ready location, and early control, of noise sources; and a space map. Visual representation of these maps are available. The pursuit of further noise reduction is represented by liberally reflected evaluation techniques, including set simulation, which permit evaluating the operating noise a compressor would generate when installed in a working set, without having to actually install it in a set.



(a)Twin-rotary compressor

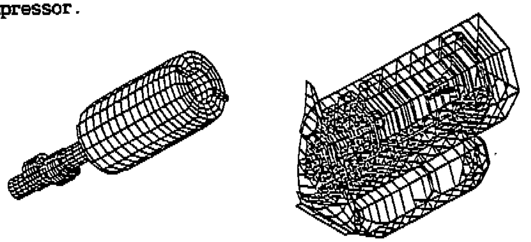
(b)single-rotary compressor

Spectrum maps of noise power intensity

Figure 6

Figure 6 shows acoustic power spectral maps of a twin rotary compressor and a single rotary compressor. The noise frequency is indicated on the horizontal axis, with the working rpm of the compressor on the vertical axis. The size of each cirle represents the power level.

Optimized designs have been achieved by the use of these evaluation technologies, coupled with structural(Figure 7) and sound field analysis of frames, mufflers, and valves. The result is 3-5 dB lower noise than a single rotary compressor.



(a)Rotor and shaft system (b)Case and frame system

Structural models of twin-rotary compressor

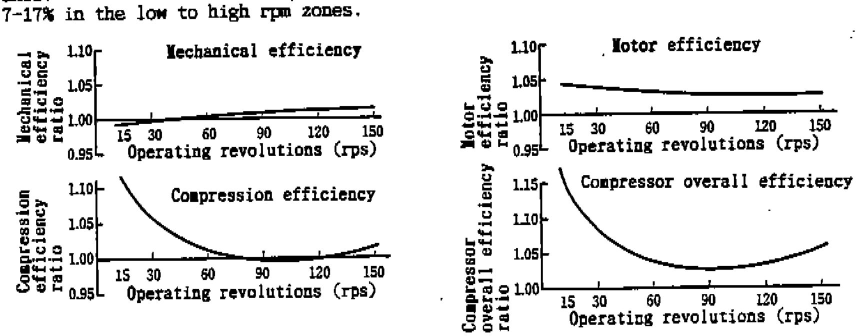
Figure 7

Better Energy Efficiency

Figure 8 compares the results of efficiency analyses of a twin rotary compressor with those of a single rotary compressor in terms of various efficiency ratios. The working rpm is taken on the horizental axis, with the mechanical efficiency, compression efficiency, motor efficiency, and overall efficiency on the vertical axis.

According to these analyses, the twin rotary compressor is a little inferior to the single rotary compressor in its mechanical efficiency resulting from local sliding losses in the low rpm zone, but surpasses it in the high rpm zone. The single rotary compressor having a smaller roller sliding area come more efficient in the low rpm zone, but gives way to the twin rotary compressor in the higher rpm zone because it offers better dynamic balancing characteristics with less shaft deformation asociated with unbalanced masses and gas pressures.

The compression efficiency, which is a factor of leakage loss, overcompression, and volumetric efficiency, has been improved in both the low and high rpm zones, reflecting cuts in the leakage loss achieved by the optimization of local sliding clearances in the compression chamber after the adoption of a new frame clamping method and manufacturing technologies, such as micron-order machining of parts and Toshiba's exclusibe self- aligning of cylinder rollers and axes. The overall efficiency, including the motor efficiency, has been improved 7-17% in the low to high rpm zones.



Analyses of compressor efficiensies

Figure 8

MECHANICAL ANALYSES

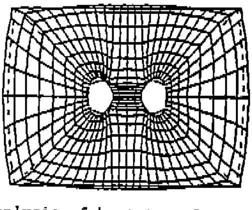
Dynamic Analysis Of Rotor-Journal Bearing System

Dynamic characteristics of the rotary shaft and bearings in the compressor are closely relatived to its performance and reliability. So far, the modes of rotor motion and bending, and bearing local characteristics have been analyzed by using bearing reactive force analyses based on the journal bearing theory, linear analyses of the rotor-journal bearing based on the finite element method, and other techniques. Linear analyses are typically conducted by using the oil film rigidty and damping coefficients determined from experimental values. These coefficients are constantly variable with the external load such that they cannot be restricted to constant values. With evolving twin rotary compressor in particular,the traditional linear equation were short of yielding satisfactory results.

As an approach to this difficulty, the authors caluculated the bearing reactive force directly with the Reynolds equations during rotor analysis so as to dynamically analyze the behavior of bending deformation of the rotor and shaft during whirling and the bearing load characteristics. By caluculation, the elastic shaft was discretized by the finite element method to determine to establish nonlinear equation of motion coupled with a short-axis apprpximate Reynolds equations by numeric analysis.

This analysis offers insight to into reactive forces from the main bearing and sub bearing, oil film pressure distributions, the orbit of the shaft center, the mode of shaft bending, and whirling of the root top (Figure 4) all a time. Optimal design of the spiral flute positions for oiling, shaft clearance, balance weight, and bearing rigidity were accomplished by using this analysis program for added bearing performance and reliability.

Thermal Stress Analysis



Analysis of heat transformation

Figure 9

While a dual-suction structure is used to suck refrigerant gas as mentioned earlier, the process of brazing pipes to adjacent holes in a deep drawn case is liable to deformation and strain under thermal stress during production. The scope of analysis by the finite element method has therefore been expand to cover the area of fitness to production, including thermal stress analysis of case hole area(Figure 9) to select optimal heating conditions, and thus to help stabilize product quality.

CONCLUSIONS

An inverter-drive twin rotary compressor has been developed which provides smooth, quiet and high-efficiency operation from low to high rpm ranges to address the user needs for better-quality air conditioning comfort, less energy bills, and less noise.

(1) The minimum working rpm has been lowered to 12 Hz to achieve a five- to ten-fold more increase in the variable working rpm range.

(2) The overall compressor efficiency has been improved 7-17%.

(3) Operating vibration has been reduced to one-thirteenth of that of a traditional single-cylinder rotary compressor, and operating noise has been cut 3-5 dB.

Twin rotary compressor are expected to open a way for greater air conditioning comfort. The authors are committed to the continuing task of perfecting compressors at ever higher lebel of performance.