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THE STUDY OF ROLLING PISTON, ROTARY COMPRESSOR DYNAMIC BEHAVIOR WHEN STOPPING TO REDUCE NOISE AND VIBRATION LEVEL

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ABSTRACT

This study presents experimental results on the rotary crankshaft behavior, the cylinder pressure and mechanical vibrations of a rolling-piston rotary compressor after shut down of the motor. It is shown as a conclusion that the most important factor characterizing transient compressor vibrations is the instantaneous crank angle at shut down of the motor, and with the instantaneous crank angle kept within a predetermined range, the transient compressor vibrations and the associated noise generation are attenuated whenever the compressor is stopped, thus permitting a more stable shut-down process of the compressor.

INTRODUCTION

Since air conditioners are often operated in quiet surroundings and are frequently switched on and off depending on the necessity, the vibration and noise generation of the compressor, both during steady operation and at shut down, must be minimized. Recently, the rolling-piston rotary compressors are used in most low capacity air conditioners, due to its high volumetric and mechanical efficiency, coupled with its compact, lightweight design. At the previous Purdue Compressor Technology Conference, the dynamic analysis of the dynamic behavior of a rolling-piston rotary compressor was presented [1]. In the present study, the transient dynamic behavior of the compressor after shut down of the electric motor is examined.

The dynamic behavior of a conventional reciprocating compressor which was used, up to several years ago, in most low capacity air conditioners, has been carefully examined by several studies [2-9]. Some of the studies derived an invention entitled "Reciprocating Compressor Having a Cut-Off Device Operable within Predetermined Angular Range," which was registered as a patent in United States of America, Japan and Australia [10], by K. Imasu, chief engineer of Matsushita Electric Industrial Co. Ltd., in 1979. The invention relates to a compressor and more practically, to the shut-down control of an electrically driven motor-compressor in which transient vibration of the compressor components and motor due to inertia force after shut down of the motor is kept to a minimal level. In the conventional reciprocating compressors, commonly, a motor-compressor unit is spring-suspended within a closed housing to form a vibration-absorbing spring system. It is quite natural that whenever the compressor is stopped, fairly large amplitude vibrations tend to take place, thus giving rise to abnormal noises, mainly due to spring-suspended compressor components striking against surrounding structures in the housing. In a conventional design, accordingly, the vibration attenuating effect during steady operation of the compressor has to be sacrificed to a certain extent, such as an increase of the spring constant for preventing large amplitude vibrations after shut down of the motor, or sufficient space resulting in a higher cost and larger size of the compressor. To be provided to prevent the compressor components from striking against the surrounding structures. In order to solve these disadvantages, the transient compressor vibrations after shut down of the motor were carefully examined. As a result, it was revealed that they are characterized by the instantaneous crank angle at shut down of the motor, and a shut-down control system of the compressor in which the motor is shut down within a predetermined angular range is fairly effective for minimizing the transient compressor vibrations.

Since the rolling-piston, rotary compressor has few reciprocation elements, the vibrations at shut down are fairly improved. However, strictly speaking, an unbalanced inertia force about the crankshaft vibrates a spring-suspended closed housing in which a compressor unit is fixed. Therefore, it may be
said that the rotary compressor also has more or less the same disadvantages as the conventional reciprocating compressors. The essential purpose of the present study is to examine whether or not the same device as the shut-down control system of the reciprocating compressors is applicable for the rotary compressor. For this purpose, transient vibrations of the rotary compressor after shut-down of the motor are measured. It is shown as a result that they exhibit a remarkable regularity, depending on the instantaneous crank angle at shut-down of the motor, thus permitting stable shut-down with the same device as the shut-down control system of the reciprocating compressors. The second purpose of this study is to reveal what rotary crankshaft behavior induces such regular transient vibrations. For this purpose, the rotary crankshaft behavior and the cylinder pressure after shut down of the motor are measured in detail. Since it is seen from the properties of the transient vibrations that a vibration amplitude increases after the first reverse rotation of the crankshaft, special attention is paid to the reverse rotation behavior of the crankshaft, thus examining the reverse rotation crank angles, their elapsed time from shut down of the motor and the maximum reverse rotation speeds. Furthermore, on the basis of measured cylinder pressure and some results of a steady state analysis on compressor vibration [11], the reverse rotation behavior of the crankshaft is discussed in detail.

Fig. 1 shows a cross section of a single-cylinder refrigerant compressor of the rolling-piston rotary type (refrigerating capacity 1755 kcal/h), which was chosen as the subject of this study. Fig. 2 shows a general view of the closed housing in which the motor and the compressor components are fixed. The housing is suspended mainly with three coiled springs. The mass of the whole compressor is 8.7 kg and the diameter of the housing is 110 mm; its length is 212 mm. The vertical crankshaft is secured at its upper portion to the motor rotor. The motor power is 0.55 kW and the average operating speed is 57 Hz when the compressor operates under load of the suction mean pressure 0.37 MPa and the discharge pressure 1.55 MPa. The refrigerant R-22 is drawn into the cylinder through an accumulator. The compressed refrigerant is discharged in the housing and so the housing inside is under high pressure. The high pressure refrigerant in the housing is transferred to a condenser through the discharge pipe attached to the top of the housing. Fig. 3 shows an underside view of the A-A' cutting plane shown in Fig. 1. The bore of the cylinder is 39 mm; its depth is 28 mm, and the outside diameter of the rolling-piston is 32.5; its eccentricity is 3.26 mm. The cylinder chamber is divided into the suction chamber (downside) and the compression chamber (upside) by a reciprocating blade with thickness of 3.2 mm. The blade tip is pushed on the rolling-piston by S-spring force and higher gas pressure in the housing. The suction port has no valve and the discharge port has a leaf valve.

COMPRESSOR VIBRATION AFTER SHUT DOWN OF MOTOR

Fig. 4 shows an outline of a shut-down control system, by which the motor is shut down at a desired rotating crank angle. Firstly, an analog signal \( V_f \) which synchronizes precisely with rotation of the crankshaft is obtained by processing a vibratory acceleration \( V_a \) of the compressor by a low-pass filter. The signal \( V_f \) is processed by a signal conditioner to obtain a pulse signal \( V_{sc} \). On the basis of this pulse signal, a synchronizing circuit provides a signal \( V_s \), by which a delay time \( \Delta t \) is added.
from a pulse signal of \( V_{sc} \) to shut down of the power source \( V_p \) is determined. The pulse signal synchronizes with a certain rotating crank angle which can be calculated on the basis of a higher-frequency vibration response in \( V_{ac} \). Therefore, the shut down of the power source can be set at a desired crank angle, by adjusting the delay time \( \delta t \).

Since the rotary compressor has few reciprocating elements, a predominant component of the compressor vibrations is the rotary vibration about the crankshaft. Fig. 5 shows the rotary vibration response after shut down of the motor, which was measured at the point B shown in Fig. 1, on the housing outside surface. The switching-off crank angle \( \theta_{cr} \) is \(-177^\circ\) in (a), \(-27^\circ\) in (b) and \(+35^\circ\) in (c). The zero degree of \( \theta_{cr} \) represents the top dead center. Each diagram shows two analogue data of the vibratory acceleration detected by an accelerometer (B & K Type 4344) and the displacement processed by a conditioning amplifier (B & K Type 2626). The maximum amplitude of such a maximum amplitude, depending on the switching-off crank angle \( \theta_{cr} \). The maximum amplitude may be classified grossly into two groups; an intense vibration between \(-260^\circ\) and \(-80^\circ\) and a weak vibration between \(-80^\circ\) and \(+100^\circ\).

**ROTARY CRANKSHAFT BEHAVIOR AFTER SHUT DOWN OF MOTOR**

In order to reveal what rotary crankshaft behavior causes the compressor vibration properties shown in Fig. 6, a flanged housing which permits measurement of the crankshaft rotation and the associated cylinder pressure was made, as shown in Fig. 7. The mass of added flange is 10 kg, and so the inertia moment of the whole compressor about the crankshaft axis is \( 36.6 \text{ N·cm}^2 \) which is larger by \( 26.6 \text{ N·cm}^2 \) than the whole compressor without flange. Fig. 8 is an upside view of the motor which shows a device to measure the crankshaft rotation. A thin aluminum disk with 72 slits is attached on the motor rotor, and a photo-coupler comprising a photo-transistor and a photo-diode is fixed on the housing inside. Fig. 9 is a downside view of the compressor, in which the two quartz pressure transducers (Kisler, 601A) for detecting the compression-side and suction-side pressure are attached to the lower crank journal. Two pressure holes of 1 mm which lead to the suction chamber and the compression chamber respectively, are made in the lower crank journal just near the reciprocating blade. The length of the pressure holes between the cylinder and the pressure transducer is...
Fig. 11 Three main patterns of rotary crankshaft behavior after shut down of motor

Fig. 12 Turning crank angle and angular velocity of crankshaft after shut down of motor, which were calculated from crank-pulses analogue data

The switching crank angle \( \theta_{cr} \) was \(-157.5^\circ\) in (a), \(-10.0^\circ\) in (b) and \(+67.5^\circ\) in (c). The abscissa is the elapsed time. The time at which the motor was shut down is known from waves of the electric input current and is indicated by zero on the abscissa. The energy in the rotating crankshaft at that instant is consumed in compressing the refrigerant, mechanical friction, oil pump, and gas leakage from piston or blade clearance, in three or four revolutions of the crankshaft. In the diagram (a): \( \theta_{cr} = -157.5^\circ \), after 103.4 ms has elapsed, the crankshaft can not pass the top dead center and the rolling-piston rotary mechanism temporarily stops at \( \theta = 220^\circ \) in the compression stroke. At that time, the gas force in the cylinder and the blade force pushing the rolling-piston work to revolve the crankshaft in the reverse direction, and so the crankshaft starts to make reverse rotation. The maximum speed of reverse rotation \( |\theta_{max}| \) reaches 87.3 rad/s (13.9 Hz), as seen from (a) in Fig. 12, and interestingly enough, the crankshaft passes the top dead center, after 166.6 ms has elapsed. This passing is known from "T.D.C." signal on the crank pulses, and from a slight increase of the suction-side cylinder pressure which is indicated by \( P_s \). The first reverse rotation continues during 107 ms. The second reverse rotation arises at \( \theta = 220^\circ \), after 212 ms; the third at \( \theta = 275^\circ \), after 271 ms; and the forth at \( \theta = 230^\circ \), after 329 ms. Lastly, the crankshaft stops at \( \theta = 240^\circ \), after 446 ms. Whenever the crankshaft starts to make a reverse rotation, the vibratory acceleration: the forth curve in Fig. 11, provides the impulsive responses indicated by \( V_1 \), \( V_2 \), and \( V_3 \).
The time between 97 ms and 125 ms has elapsed (see Fig. 13), these figures. Each group shows the following patterns of the crankshaft behavior, the main factors representing the reverse rotation behavior of the crankshaft are the elapsed time and the crank angle at which the crankshaft starts to make a reverse rotation at \( 9-260^\circ \), after 86.2 ms. Hence, the reverse rotation is so intense that the crankshaft can pass the top dead center (0 deg. of the ordinate), as seen from Fig. 14. The second reverse rotation arises at the crank angle between \(-105^\circ \) and \(-140^\circ \). The continued time from the first reverse rotation to the second is between 108 ms and 169 ms, as seen from Fig. 13. After that, as shown in Fig. 15, the maximum speed of the reverse rotation of the second reverse rotation a smaller value between 21.0 rad/s (3.3 Hz) and 61.2 rad/s (9.7 Hz), compared with the values in the intense reverse rotation group. Hence, the reverse rotation is so weak that the crankshaft cannot pass the top dead center, as seen from Fig. 14. The second reverse rotation at the crank angle between 115° and 180°.

As seen from Fig. 13, the continued time of the first reverse rotation is about 52 ms and it is less than the half of the continued time (108 ms to 169 ms) in the intense reverse rotation group. After that, the crankshaft makes the third reverse rotation at the crank angle between 200° and 235° and lastly stops near the crank angle, as shown in Fig. 14. The elapsing time of the entire stop is between 234 ms and 298 ms, as shown in Fig. 15.

The sudden stop group: After the time between 84.9 ms and 107.7 ms has elapsed (see Fig. 13), the crankshaft suddenly stops at the crank angle between 310° and 360° (the top dead center), as shown in Fig. 14.

**DISCUSSION OF RESULTS**

Comparing the maximum speed of the reverse rotation of the crankshaft which was shown in Fig. 15 with the maximum vibration amplitude of the compressor which
Hence, the loading torque \( L_{bm} \) due to the cylinder pressure is given by:

\[
L_{bm} = -eF_s \sin(\theta + \varepsilon)
\]

Subsequently, an interest should be taken in what mechanism induced such regular reverse rotation behavior of the crankshaft. In order to explain the mechanism, Fig. 16 shows some results of a steady state analysis on compressor vibration [1]: the loading torque due to the cylinder pressure, due to the blade motion, and due to the crank-journal friction. The resultant cylinder pressure force \( F_p \) and the blade force \( F_{vn} \), which act on the rolling-piston, are shown in Fig. 17. The direction of \( F_p \) passes on the rolling-piston center \( O_p \) from the compression-side chamber (right) to the suction-side chamber (left) and is perpendicular to the secant \( AB \) of the rolling-piston. Hence, the loading torque \( L_{cp} \) due to the cylinder pressure is given by:

\[
L_{cp} = eF_s \sin(\theta + \varepsilon) / 2
\]

in which \( e \) is the eccentricity of \( O_p \) from the cylinder center, \( \theta \) is the turning crank angle of the rolling-piston from the blade center axis \( x \) in the counter-clockwise direction, and \( \varepsilon \) is the turning crank angle of the line joining \( O_p \) and the blade tip center \( O_t \) from the \( x \)-axis in the clockwise direction. \( F_p \) is entirely determined by the measured cylinder pressure shown in Fig. 10. The change of the suction-side cylinder pressure is very small, compared with the larger change of the compression-side cylinder pressure. Hence, the loading torque \( L_{cp} \) changes, mainly depending on the compression-side cylinder pressure. The calculated result is shown in Fig. 16. The torque \( L_{cp} \) increases from zero, as the crankshaft turns. The steep peak of 3.5 Nm appears at the crank angle \( 0=220^\circ \), and after that, the torque decreases to zero. This steep peak corresponds to the highest peak of the compression-side cylinder pressure shown in Fig. 10. Therefore, after this steep peak has appeared, the compressed refrigerant gas in the cylinder is discharged. The main force component which induces the loading torque due to the blade motion is the blade force \( F_{vn} \) (see Fig. 17), and its direction naturally passes on \( O_p \). Hence, the loading torque \( L_{bm} \) due to the blade motion is given by:

\[
L_{bm} = -eF_s \sin(\theta + \varepsilon)
\]

\( F_{vn} \) can be given by solving an exact equation of motion of the rolling-piston crank mechanism, and the previous study [11] has shown its calculated result (ref. Fig. 14 in [11]). On the basis of the result, the torque \( L_{bm} \) shown in Fig. 16 can be obtained. \( L_{bm} \) behaves like a negative sine wave, as seen also from the expression (2): At the first half cycle, the torque has a negative value which means that the blade force induces the driving torque; at the last half cycle, the torque has a positive value which means that the blade force induces the loading torque. The maximal value of \( L_{bm} \) is \( 3.4 \) Nm. The loading torque \( L_{cjf} \) due to the crank-journal friction, which is shown in Fig. 16, can be calculated as follows: When the resultant cylinder pressure force \( F_p \) and the blade force \( F_{vn} \) are calculated (ref. Fig. 14 in [11]). Furthermore, it may be concluded that the reverse rotation behavior of the crankshaft, which possesses a high regularity shown in Figs. 13, 14 and 15, induced such regular reverse rotation behavior.

It is naturally considered that those results shown in Fig. 16 represent roughly the torque acting on the crankshaft when it starts to make the first
reverse rotation. Hence, the following significant information on the first reverse rotation behavior of the crankshaft can be obtained from those results.

The intense reverse rotation group: The group has the first reverse rotation crank-angle region between 190° and 240° (see Fig. 14). Its crank-angle region is indicated in Fig. 16. In this region, the torque Lcp due to the cylinder pressure is very large, and the potential energy stored in the compressed refrigerant in the cylinder is also large, since no amount of the compressed refrigerant has been discharged or amount of the discharged refrigerant is very small. This large torque induces the intense reverse rotation of the crankshaft, and the stored large potential energy keeps its intense reverse rotation. Of course, the torque Lbm due to the blade motion works to help the reverse rotation, and the torque Lcjf due to the crank-journal friction works to prevent the crankshaft from making the reverse rotation. After the crankshaft has passed the lower dead center (θ=180°), the torque Lcjf decreases as small as negligible, and the torque Lbm works to prevent the crankshaft from making the reverse rotation.

The weak reverse rotation group: The first reverse rotation crank-angle region (240°<θ<310°) is indicated in Fig. 16. In this region, the torque Lcp is smaller than the previous group, except a narrow region near θ=240°. Moreover, since large amount of the compressed refrigerant in the cylinder has been discharged, the potential energy stored in the compressed refrigerant in the cylinder is smaller than the previous group. Therefore, the crankshaft can not make the reverse rotation so intensively as the previous group.

The sudden stop group: The first reverse rotation crank-angle region (310°<θ<360°) is indicated in Fig. 16. When the crankshaft temporarily stops before top dead center, the frictional state naturally examined in the previous study [6] that the coefficient of the static friction in refrigerant compressors takes about 0.16. So, the frictional torque due to the crank-journal static friction at the temporal stop shows a value four times as much as the value of Lcjf shown in Fig. 16. Therefore, the sum of the torque Lcp and Lbm cannot overcome the frictional torque at temporal stop, thus permitting the sudden stop of the mechanism.

Next, close observation of the cylinder pressure after the first reverse rotation has started, is helpful for understanding the crankshaft behavior toward the second reverse rotation. The analogue curves of the cylinder pressure has been shown in Fig. 11. The suction-side pressure is almost constant during reverse rotation, because the suction port has no valve. The slow change of the analogue curves has no meaning, since it was mainly caused by a temperature change. On comparing the two compression-side pressure curves shown in the diagrams (a) and (b) of Fig. 11, there exists a great difference between them. The cylinder pressure P1 at the beginning of the first reverse rotation is almost same between them and nearly equal to the discharge mean pressure (1.5 MPa), of which values are plotted at the abscissa θ=−157.5° and −10.0° of Fig. 18 by the sign ●. As the crankshaft turns reversely, the pressure P1 decreases. The pressure P1 in (a) of Fig. 11 approaches P2, which is nearly equal to the suction mean pressure (0.37 MPa), as the crankshaft rotates toward the top dead center. Hence, the cylinder pressure does not become lower than the suction mean pressure, before the crankshaft passes the top dead center. After the crankshaft has passed the top dead center, the torque Lbm due to the blade force accelerates the crankshaft. Then, the cylinder pressure decreases from the suction mean pressure to P2 = 0.13 MPa at the second reverse rotation. On the other hand, P1 in (b) has decreased to the suction mean pressure at the crank angle θ about 180°, and after that, the pressure decreases furthermore to P2 = 0.17 MPa. The values of P1 and P2 are plotted in Fig. 18, by the sign o and ▲, respectively. From these facts, it is evident that in the case (a), the cylinder pressure works to continuously help the reverse rotation until the crankshaft passes the top dead center, thus permitting passage of the top dead center; and on the contrary, in the case (b), the cylinder pressure works to prevent the crankshaft from carrying on the reverse rotation after the crankshaft has turned to a certain extent, thus making the second reverse rotation before passing the top dead center. It is clear that the difference of the pressure properties described above naturally depends on a foregoing difference of the potential energy stored in the compressed refrigerant in the cylinder, at temporal stop. All data examined on the cylinder-pressure properties are plotted in Fig. 18. The data in the intense reverse rotation group show the similar tendency with the case (a) of Fig. 11, and the data in the weak reverse rotation group also show the similar tendency with the case (b).

Even when the steady operating conditions of the compressor: the discharge mean pressure and the suction mean pressure, are different from those in this study, the essential properties of the compressor when it stops do not change. That is, the same characteristics of the first reverse-rotation crank-angle as shown in Fig. 14 will appear, of course, in a different switching-off crank-angle region. Therefore, the same discussions as described above are applicable again for the cases under different operating conditions of the compressor. In order that the function of the shut-down control system shown in Fig. 4 be most effective, some circuit detecting the operating conditions should be added to the control system.

CONCLUSIONS

This paper dealt with the dynamic behavior of a rolling-piston rotary compressor when it stops under the constant-operating conditions (the discharge mean pressure 1.55 MPa and the suction mean pressure 0.37 MPa), to reduce vibration and noise level, and it has been made clear that:

1. The rotary vibration behavior of the compressor after shut down of the motor is entirely determined by the instantaneous crank angle at shut down, and it is classified into two groups named "Intense vibration group" and "Weak vibration group."

2. The intense vibrations of the compressor are caused by the crankshaft rotation behavior after the first reverse rotation.
3 The reverse rotation behavior of the crankshaft is entirely determined by the instantaneous crank angle at shut down of the motor, and it is classified into three groups named "Intense reverse rotation group," "Weak reverse rotation group," and "Sudden stop group." The first group appears when the first reverse-rotation crank-angle is between 190° and 240°, the second between 240° and 310°, and the third between 310° and 360°.

4 When the first reverse rotation of the crankshaft starts to make, the intense reverse rotation group has a large reverse rotation torque mainly due to the compression-side cylinder pressure, and moreover has a high potential energy stored in the compressed refrigerant in the cylinder. Therefore, the reverse rotation is so intense that the crankshaft passes the top dead center. On the contrary, the weak reverse rotation group has a reverse rotation torque and a potential energy which are smaller than the first group. So, the reverse rotation is so weak that the crankshaft cannot pass the top dead center. In the sudden stop group, the reverse rotation torque is very small and so it cannot overcome a static frictional torque at the crank-journal, thus permitting a sudden stop of the crankshaft.

5 The intense reverse rotation group of the crankshaft behavior induces the intense vibration group of the compressor, and both the weak reverse rotation group and the sudden stop group result in the weak vibration group.

As concluded above, the dynamic behavior of the compressor after shut down of the motor has a high regularity depending on the instantaneous crank angle at shut down, thus permitting a stable shut-down process of the compressor by making use of a shut-down control system, for instance, presented in this study.

ACKNOWLEDGEMENT

The authors would like to express their great thanks to Mr. Michio YAMAMURA, Director of Compressor Division and Mr. Shiro YAMAMURA, Director of Air-conditioner Division, Mr. Kenichiro IMASU and Mr. Jiro YUDA, Chief Engineers of Air-conditioner Division, Matsushita Electric Industrial Co., Ltd., for their financial support in carrying out this work and their permission to publish this work. They also wish to express their sincere thanks to Mr. Masahiko TERAKUBO, Mr. Akihiko SHIMIZU, Engineers of the Compressor Division, Mr. Hiroyuki ISHIKAWA, Mr. Naoki NISHIMURA, Mr. Atsuo MURAKAMI, Mr. Noriyoshi IMAGAKI and Mr. Takashi SAITO, for their help in carrying out the experimental and theoretical work of this study.

This paper was written while N. ISHIKI is invited to the Institute of Hydromechanics, University of Karlsruhe, Federal Republic of Germany, as a Research Fellow of the Alexander von Humboldt Foundation of Federal Republic of Germany. The authors express their cordial thanks to the Alexander von Humboldt Foundation and his host Professor, Doctor Eduard NAUDASCHER, Director of Institute of Hydromechanics, University of Karlsruhe, for their helpful financial support.

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