

1980

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## HIGH FREQUENCY VIBRATORY COMPONENTS CAUSED BY ELASTIC VIBRATIONS OF THE CRANK SHAFT IN REFRIGERATING COMPRESSORS

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### ABSTRACT

When the vibrations of a single-cylinder reciprocating compressor, which arise after the electric power for the compressor is switched off, were studied, the authors observed that the comparatively high frequency damped vibration components arise very distinctly. In this paper, the frequency and the damping property of the observed high frequency damped vibration components and the exciting force which causes the vibratory components are examined. It is concluded that the source of the damped vibration components is the transversal and the torsional elastic vibrations of the crank shaft, and the elastic vibrations of the crank shaft is one of the sources of the high frequency vibratory components of the compressor which arise when it is steadily operated.

### INTRODUCTION

In air-conditioners, refrigerators, and so on which are used in quiet circumstances, it is an important problem to decrease vibrations and noises which arise from refrigerating compressors. Especially, the comparatively high frequency vibratory components cause a serious problem when transmitted to the outside of the housing. Hence, it is extremely important in vibrations and noises-proof to find the sources of the high frequency vibrations and noises. When the vibrations of a widely used and single-cylinder reciprocating compressor were studied [1,2], the authors unexpectedly noticed that the comparatively high frequency vibratory components distinctly arise when the compressor stops [3]. In this paper, the source of the high frequency vibratory components is revealed, and on the basis of the obtained result, the same high frequency vibratory components of the compressor which arise when it is steadily operated are discussed. Many studies [4] have indicated that the sources of the high frequency vibrations of compressors are the motor (motor noises), the reed valves (valve noises) and the discharged flow of the refrigerant (gas impulse noises), and so on. However, many high frequency vibratory components which are not caused by the above vibration sources can be observed in the high frequency vibrations of compressors, and all of the vibration sources of compressors are not revealed.

When the electric power for the compressor is switched off, it is observed that the crank shaft starts to make some reverse revolutions, or it occasionally stops suddenly near the top dead center, after it has revolved sometimes [5]. After the crank shaft starts to make a reverse revolution or it suddenly stops, the motor noises, the valve noises and the gas impulse noises which the previous studies have found to be the vibration sources of compressors when they are steadily operated, do not occur logically. Nevertheless, the distinct and comparatively high frequency damped vibration components are observed in the vibrations of the compressor which arise when it stops. They arise at the instant when the crank shaft starts to make a reverse revolution, or when it stops suddenly. Firstly, the authors tried to reveal an exciting force which cause their damped vibration components. Since the friction state at each pair of the piston-crank mechanism changes from the dynamic to the static friction, it was concluded that a change of the frictional force is an exciting force which causes the high frequency damped vibration components. Generally in refrigerating compressors, a comparatively heavy motor rotor is attached to one end of the comparatively long and slender crank shaft, and hence the construction of the crank shaft is elastic and it easily causes torsional and transversal elastic vibrations. Since a change of the frictional force at each pair of the mechanism is closely related to the motion of each element of the mechanism, the change will cause the elastic vibrations of the crank shaft system, and they will cause the high frequency damped vibration components observed when the compressor stops. Accordingly, the frequency response and the impulse test characteristics of the elastic vibrations of the crank shaft were obtained, and they were compared with the high frequency damped vibration components of the compressor. It was concluded that the frequency of the elastic vibrations closely agrees with that of the high frequency damped vibration components and the damping property of some damped vibration component closely agrees with that of the torsional vibration of the crank shaft.

It is expected that the above mentioned elastic vibrations of the crank shaft naturally cause the

high frequency vibratory components of the compressor which arise when it is steadily operated. Accordingly, the power spectrum of the vibrations of the compressor under steady operation was obtained, and it was compared with the frequency response characteristics of the crank shaft. It was concluded that the peaks of the power spectrum in comparatively high frequency range correspond well to the frequency of the elastic vibrations of the crank shaft.

#### SINGLE-CYLINDER RECIPROCATING COMPRESSOR

The construction of a single-cylinder reciprocating compressor chosen as the subject of the study is shown in Fig.1. The compressor is suspended with three coiled spring which are comparatively flexible and a bending tube for discharging the compressed refrigerant R-22. The whole compressor is sealed completely in the housing. The clearance of the compressor and the closed housing is about 5 to 13 mm. The size of the compressor is 275×203×122 mm and the weight 140 N. The vertical crank shaft is supported with two crank journals and it is secured at its upper portion to the motor rotor. The power of the motor is 1.1 kW and the average rotating speed of the crank shaft is 57 Hz when the compressor operates under the following standard conditions: the suction pressure is  $3.9 \times 10^5$  Pa and the discharge pressure is  $2.1 \times 10^6$  Pa. A cascade type oil pump is attached to the lower end of the crank shaft to lubricate each pair of the piston-crank mechanism. The rotating radius of the crank pin is 10.6 mm, the length of the connecting rod is 42.7 mm and the diameter of the piston is 38.8 mm. The piston has no piston ring and the piston clearance is about  $8 \sim 10 \mu\text{m}$ . The piston pin clearance is  $3 \sim 13 \mu\text{m}$ , the crank pin clearance is  $13 \sim 28 \mu\text{m}$  and the crank shaft clearance is  $25 \sim 43 \mu\text{m}$ . The valve plate has a suction and a discharge reed valve. The size of the suction reed valve is 50×16 mm and its thickness is 0.38 mm. The size of the discharge reed valve is 66×10 mm and its thickness is 0.3 mm.

The oscillation of the compressor around the crank shaft is especially great when the compressor stops. In order to measure this oscillation, an acceleration transducer is attached to the motor stator, perpendicularly to the paper, as shown in Fig.1. The position is 70 mm away from the crank shaft and 141 mm above the center of the crank pin.

#### HIGH FREQUENCY DAMPED VIBRATION COMPONENTS

Fig.2 and 3 show the behavior of the compressor observed after the electric power was switched off. The first line shows the cylinder pressure, the second the oscillatory acceleration of the compressor, the third the electric current and the fourth the pulses representing the revolutions of the crank shaft. The dense part of the pulses represents the top dead center. The pulse interval corresponds to the rotating crank angle  $5^\circ$ . The maximum cylinder pressure is  $2.7 \times 10^6$  Pa and the minimum  $3.5 \times 10^5$  Pa. The maximum oscillatory acceleration is  $+18 \text{ m/s}^2$  and the minimum  $-25 \text{ m/s}^2$ .

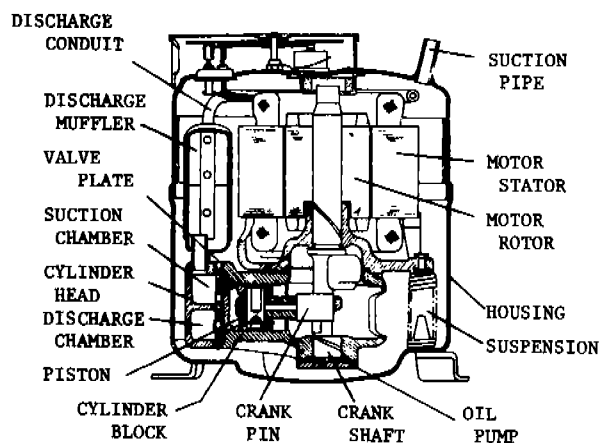


Fig.1 Construction of compressor

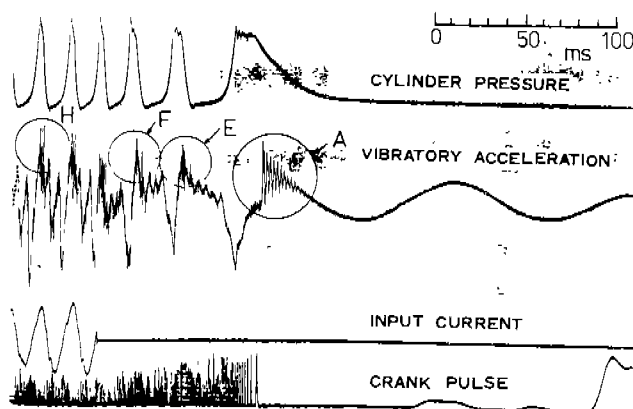


Fig.2 Vibration of compressor which arose when crank-shaft stopped suddenly

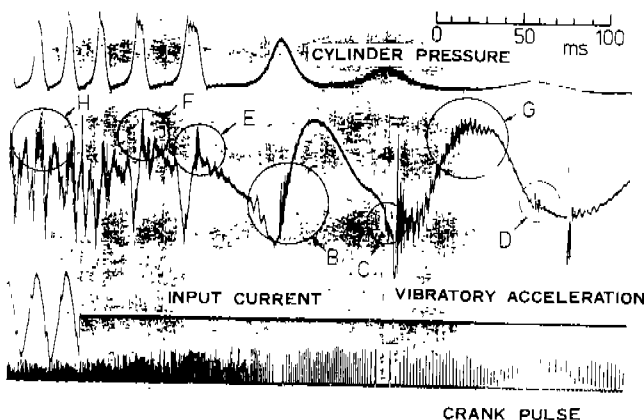


Fig.3 Vibration of compressor which arose when crank shaft made reverse revolutions

When the electric power is switched off, the energy of the rotating shaft is consumed, in the compression works of the refrigerant, the friction works of the mechanical friction, the oil pump works and the gas leakage from the piston clearance, in three or four revolutions of the crank shaft. When about 90 to 125 ms had elapsed after switching off the electric power, the crank shaft could not pass the top dead center and the piston-crank mechanism temporally stopped in the compression stroke. At this time, the gas force exerted on the piston, works to revolve the crank shaft in the reverse direction. However, when the crank angle at this time is very near the top dead center, it cannot start to make a reverse revolution [6] and it stops suddenly. Fig.2 shows one of experimental records in which the crank shaft suddenly stopped near the top dead center. The crank shaft stopped when about 93 ms had elapsed after switching off the electric power, and the crank angle was  $5^\circ$  before the top dead center. The rapid drop of the cylinder pressure after the sudden stop was caused only by leakage effects from the piston clearance [7]. When the crank angle at the instant of the temporal stop is not near the top dead center, the crank shaft can start to make a reverse revolution. Fig.3 shows one of the experimental records in which the crank shaft was able to make a reverse revolution. The crank shaft started to revolve in the reverse direction when about 122 ms had elapsed after switching off the electric power, and the crank angle was  $40^\circ$  before the top dead center.

After the crank shaft stopped suddenly or it started to make a reverse revolution, the high frequency damped vibration components as shown by "A", "B", "C" and "D" in Fig.2 and 3, which have a predominant frequency, arose. Fig.4(a) shows an enlarged diagram of "A". This damped vibration component arose at the same time as the crank shaft suddenly stopped. The predominant frequency was about 520 Hz, the maximum amplitude was about  $26.8 \text{ m/s}^2$  and the damping ratio was about 0.057. A small vibratory component of about 960 Hz was observed after this large damped vibration component. When the crank shaft suddenly stopped at the crank angle  $25^\circ$  before the top dead center, the different damped vibration as shown in Fig.4(b) was observed. A large damped vibration of which the predominant frequency was about 1400 Hz, the maximum amplitude was about  $36.9 \text{ m/s}^2$  and the damping ratio was about 0.053, arose at the sudden stop. A damped vibration of which the predominant frequency was about 520 Hz, arose after the large damped vibration, and a small vibratory component of about 1800 Hz was observed. Fig.5 shows the enlarged diagrams of "B", "C" and "D". These damped vibration components arose when the crank shaft started to make a reverse revolution. The predominant frequency of "B" was about 1800 Hz, that of "C" was about 2800 Hz and that of "D" was about 1400 Hz.

The distinct high frequency vibratory components as shown by "E" and "F" in Fig.2 and 3, arose before the crank shaft stopped suddenly or it started to make a reverse revolution. Fig.6 shows enlarged

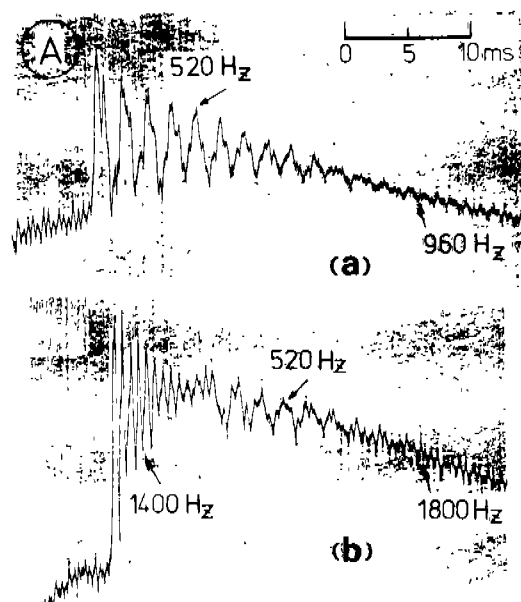


Fig.4 High frequency damped vibration components which arose when crank shaft stopped suddenly

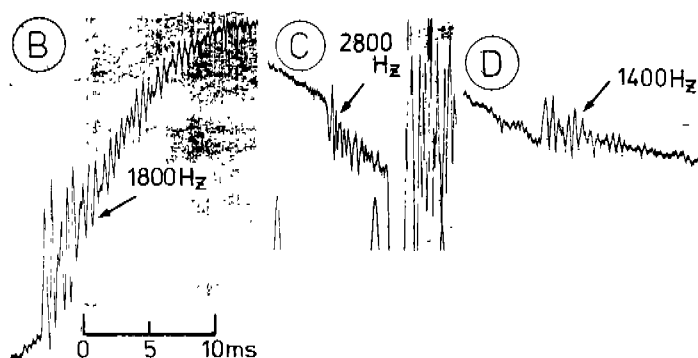


Fig.5 High frequency damped vibration components which arose when crank shaft started to make reverse revolution

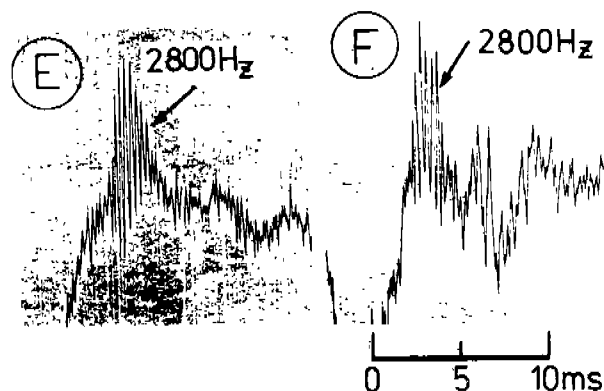


Fig.6 High frequency vibratory components which arose before temporal stop

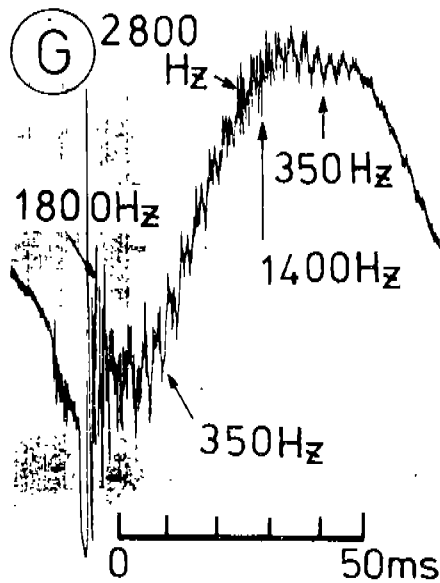


Fig.7 High frequency vibratory components which arose when compressor collided with closed housing

diagrams of "E" and "F". They arose when the crank shaft passed the top dead center and the predominant frequency was about 2800 Hz.

The rapid change of the vibratory acceleration which arose immediately after "C" in Fig.3 was caused by a collision of the compressor and the closed housing. Fig.7 shows an enlarged diagram of "E". The high frequency vibratory components of which the predominant frequencies were about 350, 1400 and 2800 Hz, were observed.

#### SOURCE OF HIGH FREQUENCY DAMPED VIBRATION COMPONENTS

##### Exciting Force

When the electric power for the compressor is switched off, the factors which cause the high frequency vibratory components are limited entirely. Firstly, the motor noises do not arise, since the electric power is switched off. Secondly, the valve noises and the gas impulse noises do not arise, especially after the crank shaft stops suddenly or it starts to make a reverse revolution. The reasons are as follows: when the crank shaft stops suddenly as shown in Fig.2, the cylinder pressure is lower than the pressure in the discharge chamber, and hence the discharge reed valve is completely closed. When the crank shaft starts to make a reverse revolution as shown in Fig.3, the cylinder pressure decreases as the crank shaft revolves and the minimum pressure is nearly equal to the pressure in the suction chamber, and hence the refrigerant is not sucked. When the crank shaft passes the bottom dead center, the cylinder pressure increases and the maximum pressure is considerably lower than the pressure in the discharge chamber, and hence the refrigerant is not discharged.

As mentioned above, the motor, the valve and the gas impulse noises revealed by many studies do not arise, especially after the crank shaft suddenly stopped or it started to make a reverse revolution. In actual facts, however, the distinct and high frequency damped vibration components arise, as shown by "A" in Fig.4 and by "B", "C" and "D" in Fig.5. These high frequency vibratory components arise when the crank shaft suddenly stops or it starts to make a reverse revolution, and they are damped vibrations. Therefore, an exciting force which causes these vibratory components arises, when the crank shaft suddenly stops or it starts to make a reverse revolution, and it is an impulsive force. The main factors which cause an impulsive force are the piston slap and a sudden change of the frictional force at each pair of the piston-crank mechanism. When the crank shaft suddenly stops or it starts to make a reverse revolution, the large gas force in the cylinder pushes the piston on one side of the cylinder wall, and hence the piston slap does not arise. The piston of the compressor has no piston ring and the piston clearance is designed very small, that is,  $8 \times 10 \mu\text{m}$ . Therefore, a large impulsive force is not caused by the piston slap, even if one arises. A change of the frictional forces arises as follows: when the crank shaft passes the top dead center, and when it stops suddenly near the top dead center or it starts to make a reverse revolution, the dynamic frictional force changes to a static one. But, in the former, the value itself of the frictional force is small, since the force which pushes the piston on the cylinder wall is small. In the latter, the value itself of the frictional forces at the piston, the piston pin, the crank pin and the crank shaft are large, and the direction of the frictional force changes. In the latter, therefore the impulsive force which acts upon each element of the mechanism is large.

When the impulsive force arises due to the change of the frictional force, it causes the elastic vibrations of each element of the compressor. An element of the compressor which is greatly affected by the impulsive force is the rotating shaft. This judgement is based on the fact that the crank shaft suddenly stops at the crank angle  $25^\circ$  before the top dead center. The reason why the crank shaft can stop suddenly is as follows: a large rotating torque is cancelled by a breaking torque due to the static frictional forces at each pair of the mechanism. Generally in refrigerating compressors, the comparatively heavy motor rotor is attached to the upper portion of the comparatively long and slender crank shaft, and hence the construction of the crank shaft is elastic and the crank shaft easily vibrates elastically. When the crank shaft stops suddenly or it starts to make a reverse revolution, the cylinder pressure is very large and this large force pushed the crank shaft and the piston, respectively against the crank journal and the cylinder wall. Hence, the elastic vibrations of the crank shaft transmit to the whole compressor through the crank journals and the cylinder wall.

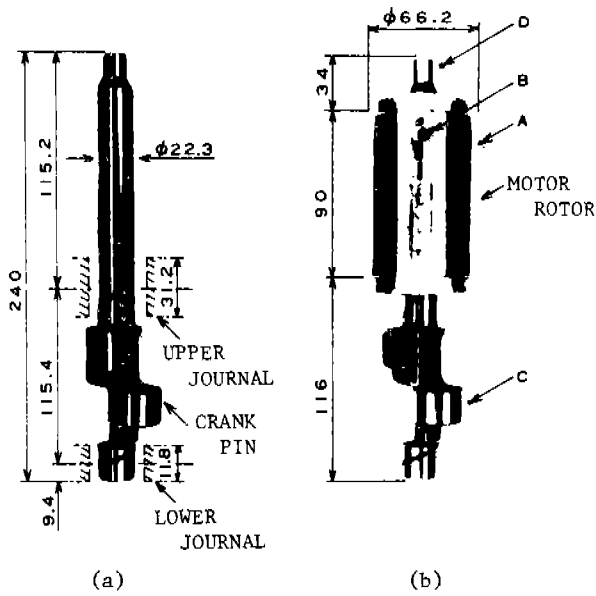


Fig. 8 Construction of crank shaft

#### Elastic Vibrations of the Crank Shaft

Fig. 8(a) shows the crank shaft in which the motor rotor is detached. The length of the shaft is 240 mm, the diameter 22.3 mm and the weight of this part 8.53 N. The crank shaft is supported by the upper journal which is 31.2 mm long and the lower one which is 11.8 mm long. Fig. 8(b) shows the crank shaft in which the motor rotor is attached to the upper portion. The motor rotor is made by piling up steel discs which have a diameter of 66.2 mm and a thickness of 0.5 mm, and the total length of the steel which is piled up is 90 mm. The weight of the motor rotor is 17.35 N. This weight is about 67 % of the total weight of the crank shaft. Hence, the construction of the crank shaft is top heavy.

Firstly, in order to examine the frequency response characteristics of the torsional vibrations of the crank shaft, the lower journal portion of the crank shaft was fixed on a base, and then point "A" shown in Fig. 8(b) was vibrated perpendicularly to the paper, and then vibratory acceleration in the transverse direction of point "B" was measured. Fig. 9(a) shows the obtained result. The predominant first resonance point of 515 Hz is closely related to the high frequency damped vibration component of 520 Hz which is shown in Fig. 4(a). Fig. 10 shows a natural oscillation of the crank shaft which was obtained by an impulse test of the torsional vibration. The predominant frequency of the damped oscillation is 515 Hz, and the damping ratio was about 0.055 of which the value closely agreed with that of the high frequency damped vibration component shown in Fig. 4(a).

Secondly, in order to examine the frequency response characteristics of the transversal vibrations of the crank shaft, both crank journal portions were fixed on a base, and then point "C" shown in Fig. 8(b) was vibrated in the transverse

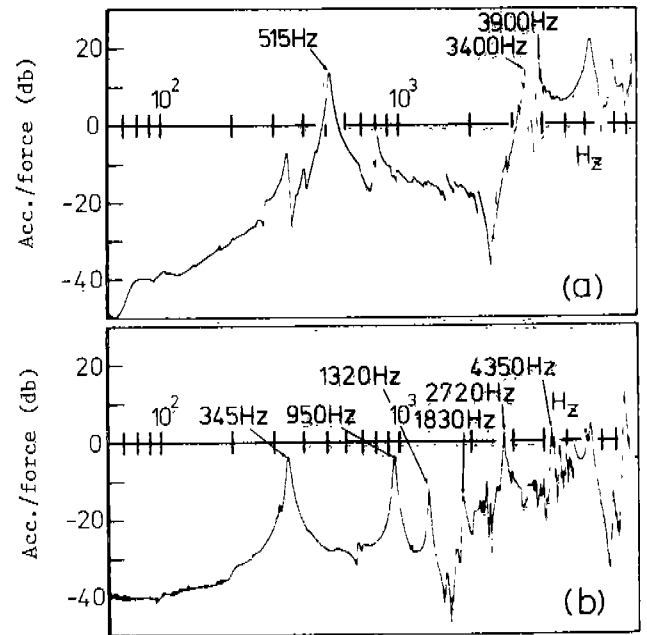


Fig. 9 Frequency response characteristics of crank shaft  
(a) torsional vibration  
(b) transversal vibration

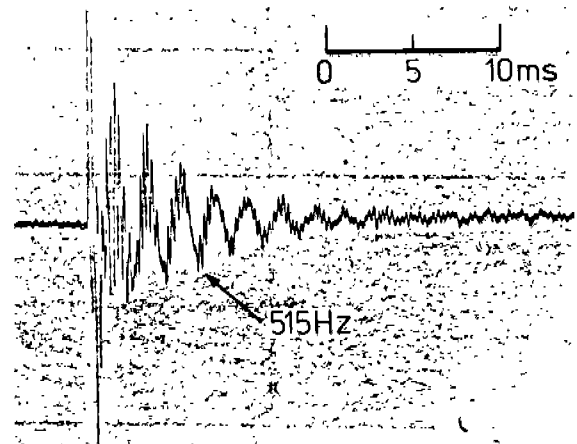


Fig. 10 Natural oscillation of crank shaft which was obtained by impulse test of torsional vibration

direction and then vibratory acceleration in the transverse direction of point "D" was measured. Fig. 9(b) shows the obtained result. The correspondence of the frequency of the revealed resonance point of the crank shaft and that of the high frequency damped vibration components of the compressor which were observed when it stops, was examined as follows: the first resonance point 345 Hz closely agrees with the damped vibration component 350 Hz shown in Fig. 7. The second 950 Hz closely agrees with 960 Hz shown in Fig. 4(a). The third 1320 Hz closely agrees with 1400 Hz which is

shown in Fig.4(b) and which is shown in "D" of Fig.5. The fourth 1830 Hz closely agrees with 1800 Hz which is shown in Fig.4(b) and which is shown in "B" of Fig.5. The fifth 2720 Hz closely agrees with 2800 Hz which is shown in Fig.6 and "C" of Fig.5.

As mentioned above, the frequency of the torsional and the transversal elastic vibrations of the crank shaft closely agreed with that of the high frequency damped vibration components of the compressor which arose when it stopped, and also the damping property of the torsional vibration of the crank shaft closely agreed with that of a high frequency damped vibration component of the compressor. Hence, it is concluded that the elastic vibrations of the crank shaft caused the high frequency damped vibration components which arose after the electric power for the compressor was switched off.

#### HIGH FREQUENCY VIBRATORY COMPONENTS OF THE COMPRESSOR UNDER A STEADY OPERATION

It is expected that the elastic vibrations of the crank shaft are one of the source of high frequency vibratory components of the compressor which arose when it was steadily operated. Fig.11 shows an enlarged diagram of "H" in Fig.2 and 3. This figure shows that the violent high frequency vibrations arose, according to a rapid decrease of the cylinder pressure after the crank shaft passed the top dead center. When these high frequency vibrations are caused by the elastic vibrations of the crank shaft, some correspondance is to be found among the power spectrum of the high frequency vibrations and the frequency response characteristics of the crank shaft. Fig.12 shows the power spectrum of the vibratory acceleration of the compressor obtained by a narrow band frequency analyzer. The power of 57 Hz is the average rotating speed of the crank shaft and the line power spectra represent the higher harmonics. Any correspondence of the power spectrum and the frequency response characteristics shown in Fig.9 cannot be found in the lower frequency range of about 1000 Hz. But, in the higher frequency range of about 1000 Hz, the peaks of the line spectra appear at 1400 Hz, 1800 Hz and 2700 Hz which closely agree with the frequency of the transversal vibrations of the crank shaft shown in Fig.9(b). Furthermore, the line spectra peak in the order of 3000 Hz has a tendency to agree with the resonance points observed as the torsional vibration of the crank shaft, and the line spectra peak in the order of 4000 Hz has a tendency to agree with the resonance points observed as the transversal vibration.

As mentioned above, the line spectra peaks in the higher frequency range of about 1000 Hz correspond well with the resonance points of the elastic vibrations of the crank shaft. Hence, these high frequency vibratory components should be called "the crank shaft noises", which are caused by a new vibration source "the crank shaft".

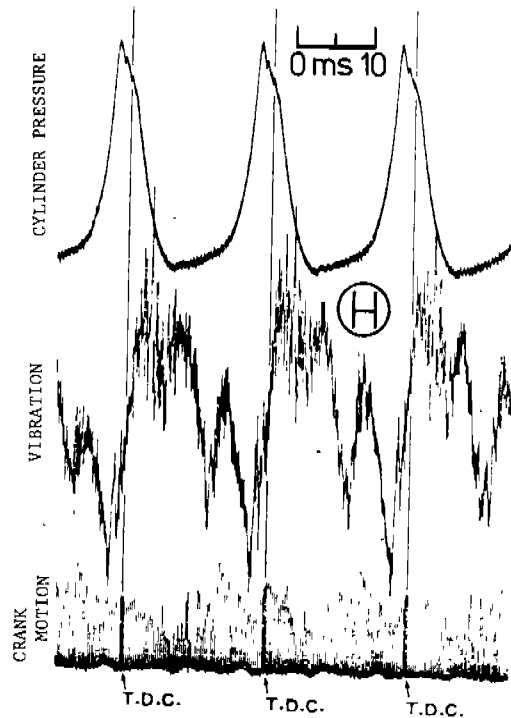


Fig.11 High frequency vibratory components of compressor which arose when it was operated steadily

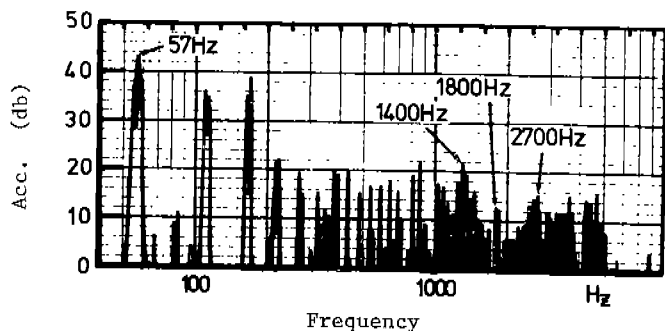


Fig.12 Power spectrum of vibratory acceleration of compressor which arose when it was operated steadily

#### CONCLUSION

Many studies have been carried out concerning the sources of high frequency vibratory components which arise in refrigerating compressors. The motor, the reed valves and the discharged gas flow have been found to be major vibration sources. However, when the crank shaft made a reverse revolution or it suddenly stopped after the electric power for the compressor was switched off, many high frequency damped vibration components arose and it was not possible to explain them under the above mentioned vibration sources. This paper

dealt with this problem especially, and it was found that there existed a new vibration source that is "the crank shaft", and the high frequency vibratory components of the compressor which arose when it was operated steadily were influenced considerably by this new vibration source. Hence, it was concluded that the high frequency vibratory components which were caused by this new vibration source, should be called "the crank shaft noises". Since an exciting force which caused the high frequency damped vibration components of the compressor which arose when it stopped, was a simple and impulsive force which was based on the rapid change of the frictional forces, the high frequency vibratory components were observed as a form of damped vibrations. Furthermore, since the damped vibrations had a predominant frequency and also since the form of damping was comparatively simple, it was possible to calculate precisely the values of the frequency and the damping ratio. For these reasons, it was comparatively easy to judge that the source of the high frequency vibratory components was the elastic vibrations of the crank shaft.

Generally in refrigerating compressors, the comparatively long and slender crank shaft is secured at its upper portion to the comparatively heavy motor rotor. Since the motor rotor is not a simple block of steel and it is made by piling up the thin steel plates along the crank shaft, it is considered that the modes of the transverse elastic vibrations of the crank shaft are especially complicated. Furthermore, the transversal elastic vibration of the crank shaft coupled with the torsional one, as seen in Fig.4(b), because the construction of the crank shaft is not symmetrical around the shaft center. The characteristics of the elastic vibrations of the crank shaft were experimentally revealed by the obtained frequency response property in this paper, and to theoretically analyze the complicated elastic vibrations of the crank shaft is very important for a precise and numerical simulation of the vibrations of compressors which also comprise the comparatively high frequency vibratory components.

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#### ACKNOWLEDGEMENT

The authors wish to express their gratitude to Dr. Masaharu Murai, Dr. Itsuo Mukunoki, Dr. Shiro Saito, Dr. Kikuo Sakata, Dr. Masanori Dan and Dr. Koichi Sawada.