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APPLICATION AND WORKING CHARACTERISTICS OF HTGR COMPONENTS TEST MACHINES WITH MAGNETIC BEARINGS

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Abstract

Simulated tests on a helium gas circulator and a reciprocating helium gas compressor with active magnetic bearings had been carried out to improve the High Temperature Gas Cooled Reactor (HTGR) components. A fair prospects has become clarified on application of the active magnetic bearing into the both kinds components. Instability problems in the high speed helium circulators which were experienced in the gas-bearing machines could be easily solved as expected and the friction-less compressor in the cylinder and piston also realized using the magnetic bearing, however, it was found the control stability has difficult problems in the active magnetic bearing system and were not necessarily satisfied the present applications. A lack of technical informations owing to the special situation on the active magnetic bearing makes very difficult analytical considerations on the test results and improvement of those machines.

1. INTRODUCTION

Though the main gas circulator of the High Temperature Gas Cooled Reactor (HTGR) system which uses the helium gas coolant is one of the most important component to keep its safety, the various problems had been encountered in the existent HTGR plants.[1]

Owing to low density of the helium gas, rotating speed of the main gas-circulator for HTGR must be designed more than 10,000rpm or several thousands rpm for one stage or multi-stage compressor respectively. Besides the problem on high speed driving, severe requirements on shaft sealing and lubrication must be satisfied during its long time operation. The shaft sealing mechanism employed mechanical seal and buffered gas seal had been applied in the main gas circulators of existent HTGRs. The submerged electric driven gas circulators had been used with the gas-bearing for the small scale research reactor[2] and are working in the Helium Engineering Demonstration Loop (HENDEL) in Japan Atomic Energy Research Institute(JAERI). They will be also used in the test reactor HTTR[3] just initiated to construct at JAERI.

The former type uses lubricating oil or water as the lubricant or sealing fluid and uses helium for the buffer gas, however, the pressure control of the buffer gas and the lubricant had not worked satisfactorily in the transient condition. This problem caused ingress of the liquid into the coolant gas and had caused fatal defect of the existent HTGR.

In case of the submerged type with gas bearings for the test reactors, weight of the rotor amounts to more than 150 kg including the motor rotor. Thus the dynamic balancing of them become extremely difficult and it tends to cause the half-speed whirling. This phenomenon is feared to grow into the shaft whip condition and to damage the gas bearing mechanism severely.

As the reciprocating compressor for the auxiliary system of HTGR and fuel reprocessing system should be designed to be gas tight and oil free in the cylinder, they could not help using oil free dry-piston or diaphragm type compressors in such applications. The technical problems in those compressors are such those as short life, larger size and radiation exposure resulting from required frequent maintenance.

The experimental works are being carried out on both of the high speed helium circulator and the reciprocating compressor employing the active magnetic bearing. The simulated test apparatus for the practical helium circulators and the small reciprocating compressor were prepared to accomplish these test.

The development test aims to solve problems on the gas bearing circulator and the oil free type reciprocating compressor and to know practical problems on applying the active magnetic bearing on the compressor.

The test results showed that the active magnetic bearing involves favorable characteristics which had never realized by the any conventional bearings, however, it is not always easy to realize the enough stability in the difficult mechanical condition. It was also found that previous analytical considerations on vibrational characteristics are very important for the mechanical components to realize a stable magnetic bearing control system.

2. TESTING APPARATUSES

2.1 SIMULATED GAS CIRCULATOR

Fig. 2.1 shows a sectional view of the simulated gas circulator test apparatus on both of the gas bearing helium circulators in service and predicted one for the test reactor "HTTR". Photo. 2.1 and 2.2 show appearances of the rotor and the magnetic bearing of the apparatus.

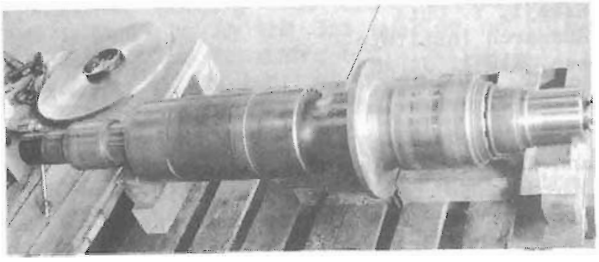

Photo. 2.1 Appearance of Simulated Gas Circulator Rotor

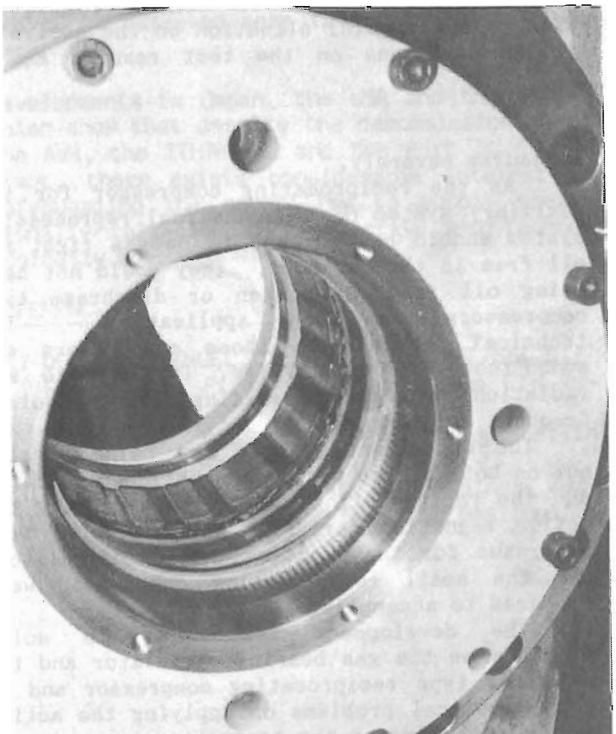

Photo. 2.2 Magnetic Bearing of Simulated Gas Circulator (Lower)

The rotor positions in vertical direction and is driven by the 30kW electric motor in the same manner as the practical circulators.

Main features of the simulated rotor are;

Total rotor length	:	1,227	mm
Max. rotor dia.	:	182	mm
Outer Dia. of journal shaft	:	169 109	mm(upper) mm(lower)
Total weight	:	172.9	kg
Moment of inertia	:	0.71	m^2kg(spin)
	:	17.6	m^2kg(transverse)

The practically important two critical speeds among the four first order flexion mode critical speeds ω_1, ω_2, ω_{b1} and ω_{b2} are estimated as 15,540rpm and 13,200rpm for forward ω_1 and backward ω_{b2} respectively.

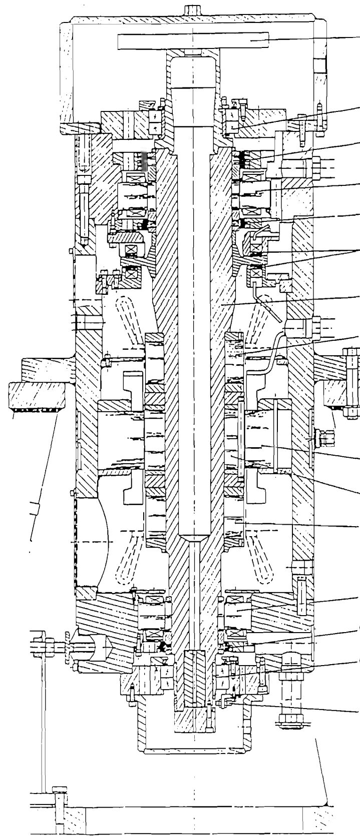

Fig. 2.1 Section of Testing Apparatus for Gas Circulator

The solid disk at upper end of the rotor simulates the impeller and the additional laminations were attached on both side of the motor rotor to simulate the larger power motor in the practical gas circulators which are estimated about 250 to 300kW. The hollow shaft saves total weight of the rotor and contributes to increase the flexion mode frequency of it.

The magnetic bearing positions at both side of the journal shaft and pair of ring shaped axial magnetic bearings are also equipped adjacent to both side of the rotating disk. Both of the journal bearings keep air gaps of 0.5mm and the axial bearing has 0.6mm gap between the rotating disk surfaces. The maximum load capacities of the journal bearings are 3,700N and 2,300N for upper and lower bearings respectively and is 7,650N for the axial bearing.

Auxiliary bearings consisted of conical ball-bearings are attached close to the magnetic journal bearings with 0.175mm radial air gap which is equivalent to 0.25mm for axial direction.

The magnets in the each bearings are excited with bias current to avoid the zero attractive force during the working condition, because when required attractive force changes across zero, the terminal voltage of the magnet burst out by the induction of the winding and the high voltage impulse might damage the amplifier.

The shaft position sensors are mounted in the levels just below the journal magnetic bearings for radial displacement and axial position is detected by pair of two detectors which positions upper side of the hollow shaft as shown in Fig. 2.1. The additional winding in the magnet detects induced magnetic flux to stabilize the attractive force causing from dependency of the distance between a magnet and the object.

Each control amplifier among five units in the control box feeds the opposite magnets in upper and lower journal bearings of two (X, Y) axes and the axial bearing of one (Z) axis. The auto-balance function is involved in the journal bearing system rejecting the rotating frequency signal component from the shaft vibration signals. This function is a unique technology which could be never realized by conventional bearings owing to the concept of Société de Mécanique Magnétique(S2M). The function makes the rotor rotate around its inertial axis causing zero-stiffness only in the rotating frequency itself. The storage batteries back up power supply system of the control box for five minutes at least.

The displacement signals are available in X, Y and Z axes for the journal bearings and the axial bearing respectively. The dynamic exciting current which is proportional to the force is also measurable for those axes. More than seventy working status signals of the control box are available through the open collector circuits and a part of them are used in this apparatus.

The semi-conductor type frequency converter changes rotor speed from nearly 0 to 12,000rpm changing the frequency from 0 to approximately 220Hz. As the rotor supported by the magnetic bearings keeps rotation without driving power for a long time, it should be rapidly decreased in the emergency condition. Thus the resistors with large heat capacities were equipped in the frequency converter to damp rotating energy of the rotor. Thereby the rotor could be stopped by the regenerative damping in a few seconds from 12,000rpm.

2.2 RECIPROCATING HELIUM COMPRESSOR

Fig. 2.2 shows cross sectional view of upper part of the testing compressor and Photo. 2.3 is an appearance of it. The compressor is being operated practically in the auxiliary circuit of the facility.

Main specifications of this compressor are as follows:

Design gas pressure	:	4.0 Mpa(inlet)
Design gas pressure rise	:	0.5 Mpa
Design gas temperature	:	30 C° (inlet)
Stroke of piston	:	35 mm
Cylinder diameter	:	85 mm
Max. rotating speed	:	1,200rpm
Design shaft power	:	7.5 kW
Weight of piston	:	≃ 1 kg
Design gas temperature	:	30 C° (inlet)
Displacement volume	:	170 lit./stroke
Equivalent mass of reciprocating part (on the level of magnetic bearing)	:	1.82 kg(B.D.C.) 2.04 kg(T.D.C.)

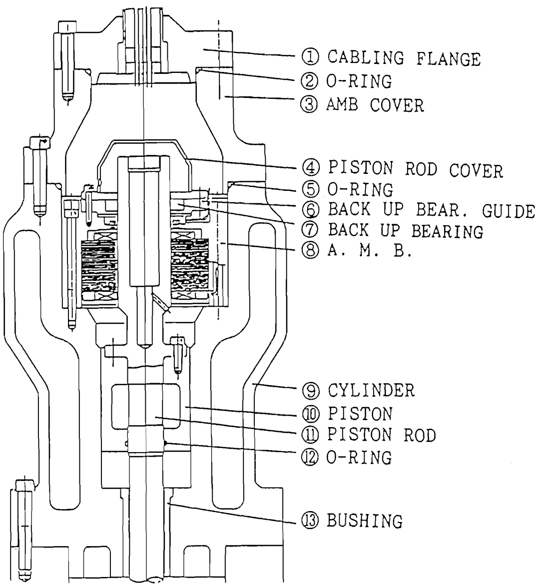

Fig.2.2 Cross Sectional View of Upper Structure of Reciprocating Compressor

Photo. 2.3 Appearance of Reciprocating Compressor

As shown above, the stroke of piston is designed very short compared to the cylinder diameter thus it makes possible to operate in higher speed.

The magnetic bearing supports the extended piston rod in radial two axes on the top of cylinder. The conventional oil lubricated cross-head supports lower end of the piston rod which fixed with it because the horizontal force component is much larger for the cross head. The oil film in the narrow clearance around the cross-head will contribute to increase the damping factor of the vibrating system. The radial clearance between the extended piston rod and the magnetic bearing is 0.35mm in normal condition. Aluminum alloy was used as a material for main part of the piston to save weight of it. Nominal clearance between labyrinthian piston and cylinder was machined into 150μm, however, this value was found to be reducible into a figure of micro-meter or less as mentioned later.

The magnetic bearing controller involves nominal electric output of 60V, 4A and equips back up batteries of five minutes working time.

An electric motor with 11kW output drives the compressor through a direct coupling and the driving speed is adjustable continuously from 0 to 1,400rpm using a frequency converter.

3. WORKING CHARACTERISTICS

3.1 SIMULATED GAS CIRCULATOR

Vibrational characteristics of the rotor shaft were measured over the whole speed rage using the shaft displacement signals. The vibrational signals were converted into time average amplitude, vibrational traces and frequency spectral maps with the rotating speed.

Fig. 3.1 shows vibrational amplitude with rotating speed of the shaft in upper journal bearing. A thin line in the Figure shows typical amplitude of the practical helium circulator with gas bearings to compare with them for the magnetic bearing. It is very interesting that the shaft amplitude for the magnetic bearing are less in general compared to gas bearing's one and are kept in a constant value in the range over 6,000rpm.

The imbalance of rotor system and stability of the control loop affect amplitude of the shaft of course, however, those values for the test machines with magnetic and gas bearings are roughly the same value 3μm to 5μm and the bending stiffness of the both rotor are almost equally designed. Thus the difference between both types might show that stiffness and damping factor of both bearings are so different. It should be remarked that the auto-balance function reduces the amplitude clearly over 6,000rpm where it works, though the amplitude in the condition must increase because the stiffness become to zero for the rotating frequency component of the vibration force. This provably means that the rotor is very well balanced, however, the control stability of the system made the amplitude relatively large in the condition where the auto-balance was working and the amplitude reduced into the original imbalance when the stiffness become to zero.

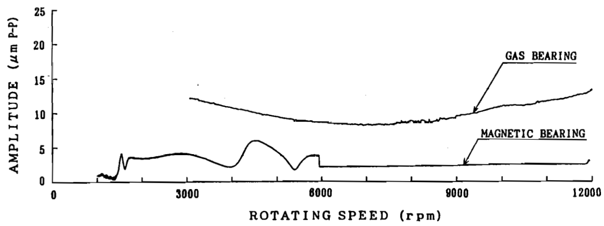

Fig. 3.1 Vibrational Amplitude with Rotating Speed (Upper Journal Bearing)

Fig. 3.2 shows vibrational spectral map measured on upper end of the journal shaft. It will be understood that the amplitude is much lower and no any higher harmonics are found in this figure compared to Fig. 3.3 which shows typical map of the practical helium circulator with gas bearings. It means the stiffness and damping factor is isogonic and the parasitic oscillation is negligible in this test machine.

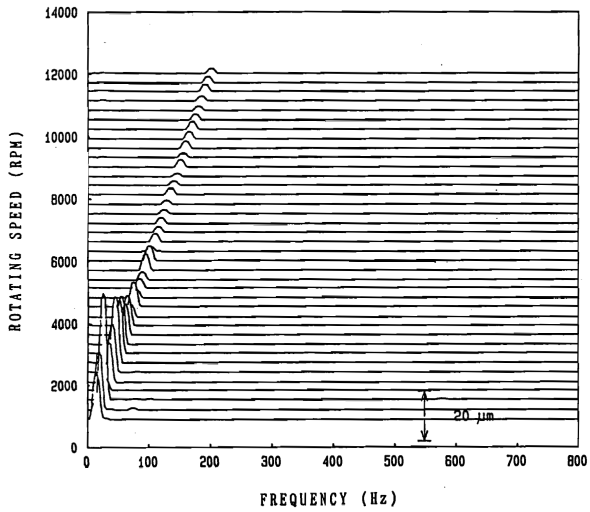

Fig. 3.2 Vibrational Spectral Map of Simulated Gas Circulator on Upper Journal

Fig. 3.4 shows typical trace of the shaft vibration measured on upper journal bearing in radial plane and with time. The almost complete circular trace shows the same characteristics described with Fig. 3.2.

The self-induced weak vibration had been observed at times in higher frequency (KHz) regime. This means stability in the higher frequency regime is insufficient, in other words, the damping factor should be increased more for the control system or steep cut down of the amplifier gain is necessary without larger phase lag in those frequency regime.

A test on the backup bearing was carried out in the rotating speed of 12,000rpm, however, no any significant problems were happened on it.

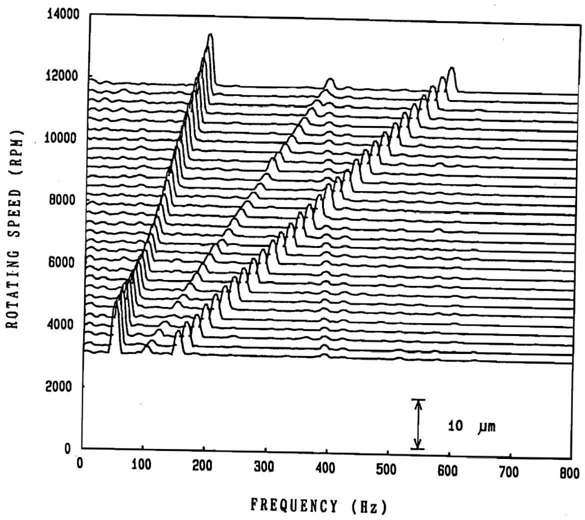

Fig. 3.3 Typical Spectral Map of Practical Helium Circulator with Gas-Bearing

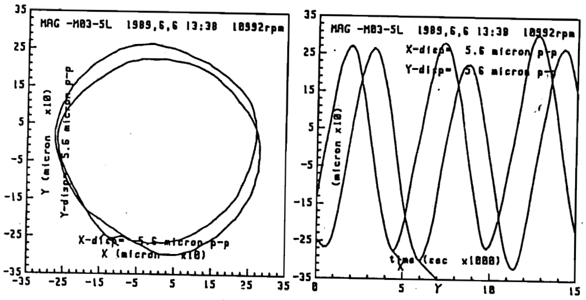

Fig. 3.4 Typical Lissajous' Figure and Time Trace of Upper Journal Vibration

3.2 RECIPROCATING HELIUM COMPRESSOR

The operating tests were accomplished with air and helium as the working gas just after the installation and the compressor have worked more than 4,600 hours on end of April 1990. The volumetric efficiency was more than 60% in rated pressure difference with helium gas and will be improved decreasing the clearance around the piston. The radial vibrational characteristics were measured in two axes using the deflection signals from the magnetic bearing.

The deflection signals were measured as time average amplitude with rotating speed and were transformed into frequency spectra in the same manner as simulated gas circulator. The vibrating trace were also plotted into Lissajous' figure and with time. Figs. 3.5 through 3.7 show typical vibrational data for the operation with helium gas.

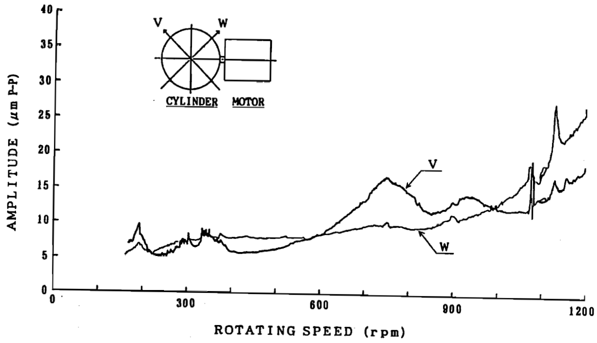

Fig. 3.5 Radial Amplitude of Piston in V and W axes

Among the figures, Fig. 3.5 shows piston amplitude in two axes of magnetic bearing V and W. The amplitude curves for two axes differ each other in general and are steeply increase over 1,100rpm. It will be seen a large peak on 750rpm for V axis, sharp peaks on around 1,100 to 1,150rpm and small peaks on less than 200rpm. The difference with two amplitude curves is thought to be mainly caused from characteristics of the magnetic bearings are not quite equal for two axes. Thus the resonant frequency and the height of peaks are unequal because they depend stiffness and damping factor of the magnetic bearings as understanding from the spectral map shown in Fig. 3.6.

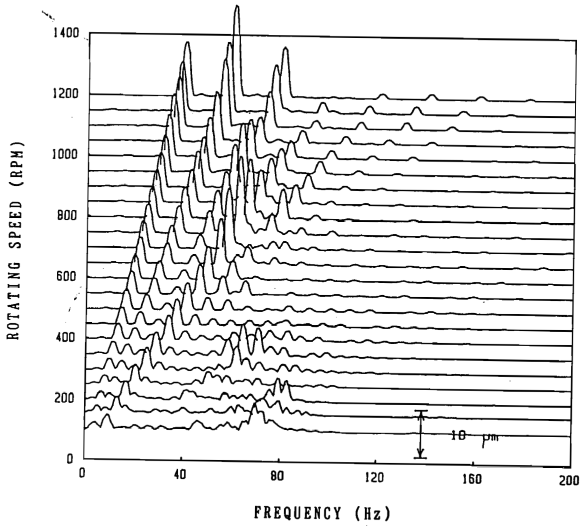

Fig. 3.6 Spectral Map of Radial Vibration of Piston

It should be remarked on Fig. 3.6 that we can find no basic frequency components of vibration but higher harmonics more than second order. These results caused from so called "Peak of Gain" which intensifies gain of the control circuits only for a narrow frequency band consisted with the rotating speed. Remarkable peaks are also found in Fig. 3.6 on vertical 60Hz line. These are thought to be parasitic vibrations and resonance of forced vibration with resonant frequency decided by stiffness of the magnetic bearing and mass of the object.

The equivalent stiffness is estimated as around 3 x 10^4 kg/m from the resonant frequency and the equivalent mass. The parasitic vibrations and resonant peaks in Figs. 3.5 and 3.6 show insufficient damping of the system. It will be realized by the increased differential gain which is consistent with advancing the phase from lower frequency in the electronic circuits. Instability will be observed from the parasitic vibration in a Lissajous' figure and a time trace for 300rpm shown in Fig. 3.7.

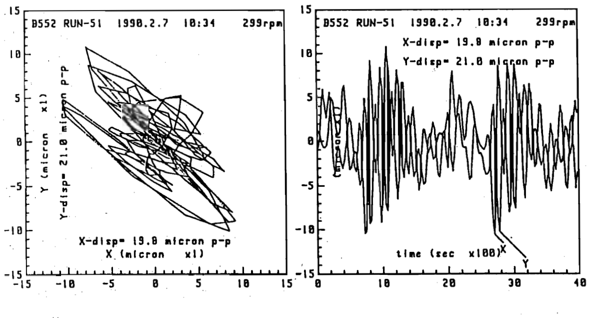

Fig. 3.7 Typical Lissajous' Figure and Time Trace of Radial Piston Vibration

In spite of the insufficient stability, the piston works in very small vibration amplitude and is kept in less than 35μm in peak to peak value. This enables to reduce the clearance around the piston from the present value of 150μm and will cause much improvement on the volumetric efficiency. Farther improvement of the stability is being discussed on dynamic characteristics of the control amplifiers.

4. DISCUSSION ON VIBRATION CHARACTERISTICS

It was found from test results on the two kinds application of the magnetic bearing that the most important technology is to realize a highly stable system. An easy way for this problem will be decreasing total gain of the system, however, it results in low stiffness and low damping factor of the magnetic bearing system which cause relatively low accuracy of control.

The phase lag between the amplifier input and deflection of the object sharply increases at the system resonant frequency in lower damping condition and causes instability of the system. Thus we should realize a large damping factor by the control circuits if the resonant frequencies are relatively low with the operating frequency. The way to solve this problem is not difficult at all in principle, that is advancing phase or enhancing differential gain of the signal for the object deflection in the circuits.

Figs. 3.8a and 3.8b show calculated frequency characteristics on amplitude response and phase respectively for the realistic constants of the compressor.

As shown in Figs. 3.8a and 3.8b, the large damping factor C_1 of the magnetic bearing suppresses the resonant peak and keeps the phase lag less than 90 degree and realizes a stable system. Though the principle way is not difficult to make a stable and accurate system by the circuits, but it is not always easy to realize in the practical circuits.

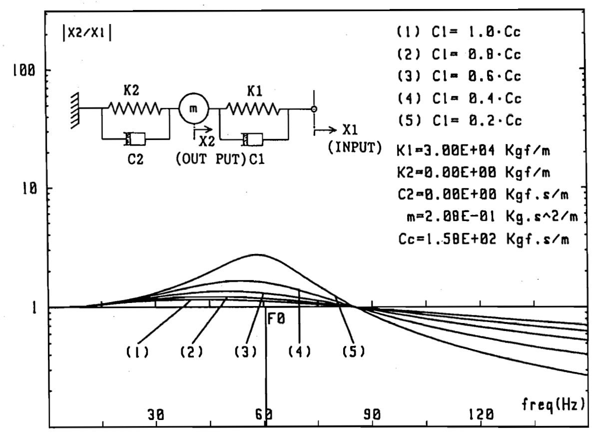

Fig. 3.8a Calculated Amplitude Response of Reciprocating Compressor

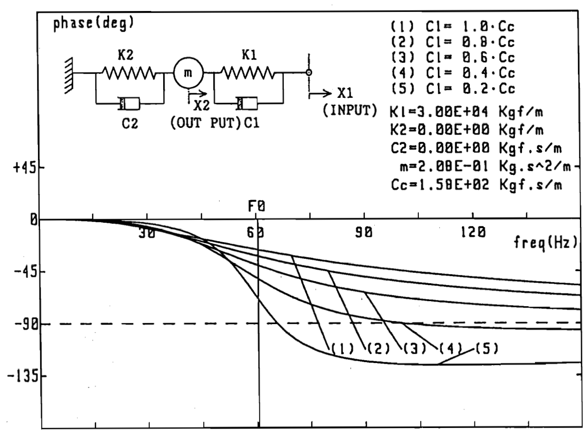

Fig. 3.8b Calculated Phase Lag of Reciprocating Compressor

Understanding the reason why, let us use a logarithmic coordinate system with horizontal frequency axis and vertical gain axis. The proportional gain which means stiffness shows a horizontal line and the differential gain shows a straight line with positive 45° inclination.

Both lines crosses at the cross-over point in the cross-over frequency. As the differential gain line shifts to upper side with the damping factor C_1, the cross-over frequency also shifts to left hand side with it. This means the required gain in higher frequency than cross-over point must be increased proportionally to the damping factor. This is apt to cause a saturation of the output for the limited input signal, furthermore, elevated gain in the higher frequency regime causes over-sensitiveness for noises and makes the electronic amplifier it self unstable.

Thus it is essential to make the imbalance or forced vibration force as small as possible to prevent saturation of the amplifier and to limit the rotational speed to avoid extremely wide frequency range for the circuits. It is so important to improve the electronic amplifier as to endure a wide signal level and as to stabilize the amplifier it self. Intensified gain in the high frequency requires the position signal with very low noises because such amplifier is very sensitive to the noises. From the same reason, electronic devices and signal lines must be carefully selected or designed to reduce the noises.

It is also important to design the mechanical components to be rigid and light weighted so that the resonant frequency could be elevated.

5. CONCLUSION

A fair prospect has become clarified on applications of the active magnetic bearing into the helium circulator and the reciprocating compressor. It was found that the instability problem on the conventional bearing has perfectly solved in the magnetic bearing machines, however, the problems on the control stability had not solved completely even for the magnetic bearing system. From the experiences on experimental studies, we point out the stability consideration is very important when the resonant frequency is relatively low for the operating speed. A lack of technical informations on the control system also makes very difficult analytical considerations on the test results and improvement of those machines owing to the special situation such as patent on the active magnetic bearing.

Nevertheless, the high speed machines and reciprocating compressors were much improved in the vibration amplitude compared to the conventional bearing system. Applying the magnetic bearings, one would have much practical merits on manufacturing, installation, maintenance and operation of the machines such as comparatively easy balancing on the rotor, simple and maintenance free bearing system, very weak vibration to the base, vibration monitoring in service and so on. The maintenance free and wearless machine will greatly contribute to minimize radiation exposure in the nuclear facilities.

The instability due to the insufficient damping factor would be subjected by the high speed digital control system which would be changeable the gain independently from the phase shift in the electronic circuits. Development of the semi-conductor devices with a large power and wide frequency range will be also important to apply the magnetic bearing into high speed and large machines.

It should be emphasized that the magnetic bearing is not only a substitute of the conventional mechanical bearings but is a innovative tool to realize the modern machines which had never produced. The uniqueness on accurate controllabilities of stiffness and damping factor and the special functions such as integral control and auto-balance should be utilized in the advanced mechanical systems.

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