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A low-vibration type compressor for refrigerators using a self-standing support method

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ABSTRACT

To reduce the vibration of reciprocating compressors used in household refrigerators, we propose a "self-standing support method" to support the drive unit in a compressor shell. We verified its validity with respect to vibration suppression effects through experiments as well as numerical calculations using a simplified model that was created based on an actual compressor structure. The results are as follows.
(1) The self-standing support method attaches a spherical support element to the bottom of the drive unit, which is directly placed on the plate. The method does not require conventional support springs, and thus it can substantially reduce the natural frequency. Therefore, the entire natural frequency can be eliminated from the region of rotational speeds for operation. We verified that the creation of resonance-caused vibration was avoided in the region of low rotational speeds.
(2) Aligning the application point of the unbalance force with the center of percussion against the contact point of the spherical support eliminates the friction force acting at the contact point even with the generation of the unbalance force, suppressing the vibration transmission from the drive unit to the plate. Accordingly, we verified that the vibration transmission to the plate could be sufficiently suppressed, even if the unbalance force acting at the drive unit increases in the region of high rotational speeds.

1. INTRODUCTION

In reciprocating compressors that are widely used in household refrigerators, the refrigerant gas in the cylinder is compressed by the reciprocating motion of a piston driven by a motor in the compressor main unit (drive unit) housed in the shell. As a significant portion of the power consumed by a refrigerator is used to drive the compressor, operating the compressor at a low rotational speed for as long as possible can reduce the power consumption (Tanaka *et al.*, 2003). Therefore, in order to maintain sufficient refrigeration performance even when operating at a low rotational speed, products with improved thermal insulation in the refrigerator body have been developed using vacuum insulation materials (Yuasa *et al.*, 2014). However, compressors currently available have a problem of vibration when operating at a low rotational speed. This is because compressors have many natural frequencies at relatively low rotational speeds, and resonance is created by the unbalanced force that is caused by the piston weight when the rotational speed of operation is closer to the natural frequency (Imaichi *et al.*, 1975). Strong vibrations in the compressor are transmitted to the refrigerator body via supporting elements, generating noise from the refrigerator. Therefore, at the region of low rotational speeds, the available region of rotational speed for operation is limited.

Therefore, in order to reduce the vibration of the compressor itself, we propose a self-standing support method to support the drive unit housed in the shell (Ike *et al.*, 2017). In this method, a block with a spherical bottom (spherical support element) is attached to the bottom of the drive unit and is directly placed within the shell. The drive unit is self-standing, with its restoring moment due to gravity. With this method, conventional coil springs used to support the drive unit are eliminated, thus reducing support stiffness, and substantially reducing the natural frequency of the drive unit. Further, with this method, the center of percussion is applied to the drive unit, which minimizes the periodic friction force acting on the contact point between the spherical support element and the shell during the operation of the compressor, thus suppressing the vibration transmission from the drive unit to the shell. This paper verifies the validity of the self-standing support method through an experiment as well as analysis of the simplified model created based on an actual compressor structure.

2. PROPOSAL OF THE SELF-STANDING SUPPORT METHOD

Figure 1 shows the schematic view of a conventional single-cylinder reciprocating compressor used in household refrigerators. This compressor mainly consists of a shell and a drive unit. The refrigerant gas flowing into the shell is compressed by the drive unit, and is externally discharged through the discharge tube. The drive unit includes a motor that rotates the crankshaft and the upper piston-crank mechanism that compresses the refrigerant gas. The drive unit is supported by the bottom of the shell via four coil springs attached underneath. To reduce natural frequencies, the spring constant of these coil springs must be reduced. However, the spring constant is usually adjusted to the minimum sufficient value necessary to support the weight of the drive unit, and therefore it is difficult to reduce it further. Therefore, this paper proposes a self-standing support method, a new method to support the drive unit.

Figure 2 shows the concept of our self-standing support method. In this method, a spherical support element instead of coil springs is installed at the bottom of the drive unit. When the center of curvature O of the sphere is located above the center of gravity G of the drive unit, the drive unit is stable at the equilibrium condition, maintaining its self-standing state by its restoring moment due to gravity. In the meantime, since the restoring moment due to gravity is smaller than that of the coil spring due to the reaction force, the supporting stiffness against the drive unit decreases compared with that of a conventional spring support method. Further, the moment of inertia around the contact point Q increases according to the distance between the center of gravity G and the contact point Q. Overall, the self-standing support method can reduce the natural frequency existing in the region of low rotational speeds of a compressor.

Further, in the drive unit supported by this method, the unbalance force generated by the reciprocating motion of the piston acts in the horizontal direction, generating a rolling vibration with the contact point Q as instantaneous center. Here, to prevent the periodical friction force from being applied on the contact point Q, the drive unit is designed to align the application point P of the unbalance force with the center of percussion against the contact point Q of the sphere. This theoretically prevents the friction force from acting on the contact point Q even if the unbalance force acts on the application point P, thus minimizing the vibration transmission to the shell.

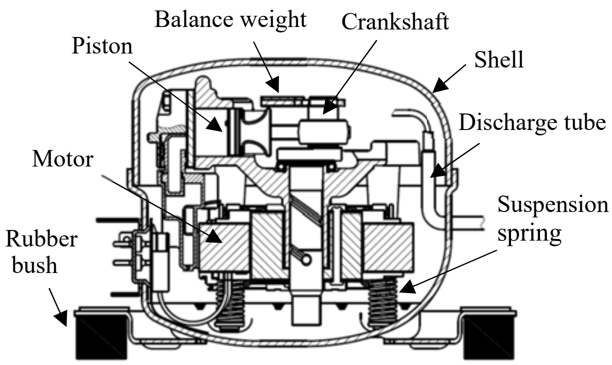


Figure 1: Schematic view of a reciprocating compressor

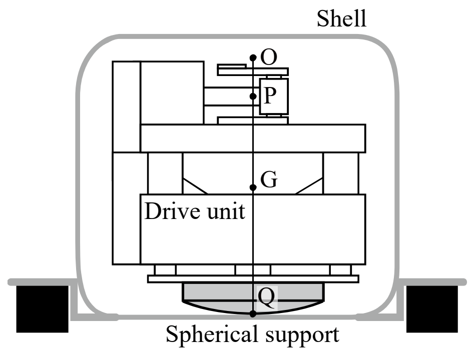
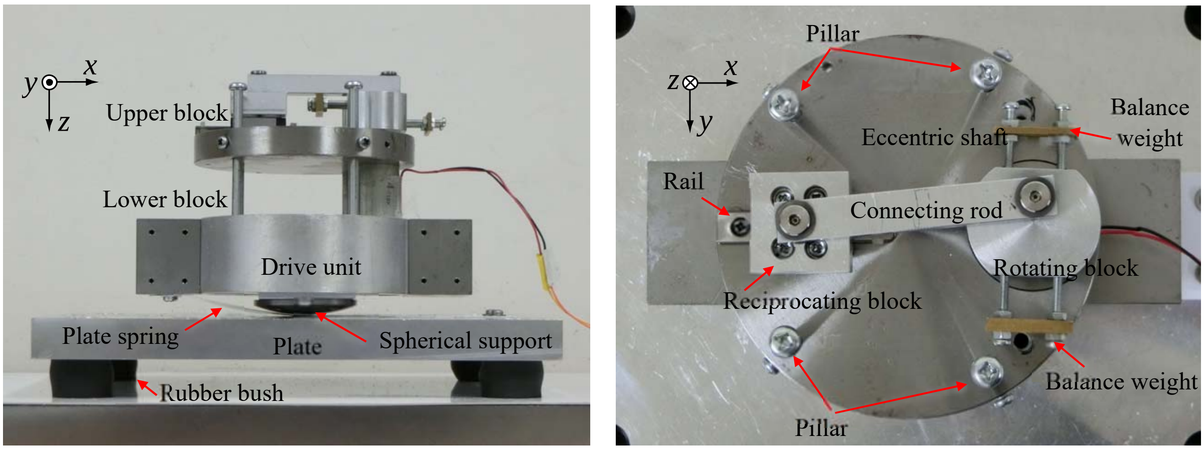


Figure 2: Concept of self-standing support



(a) Front view (b) Top view

Figure 3: Experimental apparatus (Simplified model with the self-standing support)

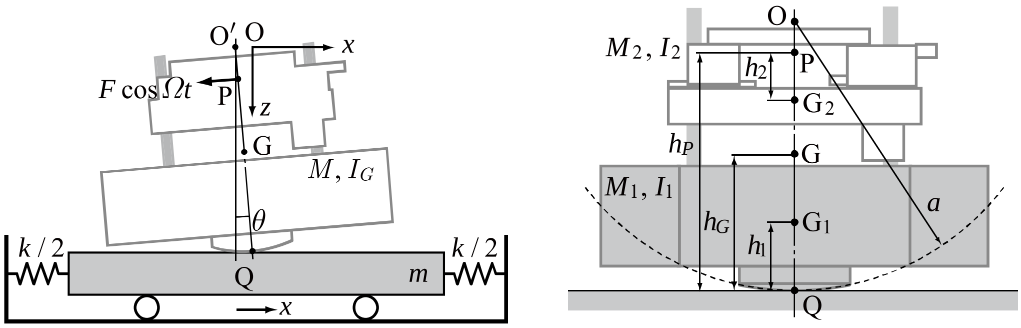
3. SIMPLIFIED MODEL

3.1 Experimental apparatus

This paper verifies the validity of the self-standing support method by using the experimental apparatus shown in Figure 3. The experimental apparatus is a simplified model, which extracts the elements necessary for verification from an actual compressor structure. The drive unit consists of an upper and a lower rigid block which are connected through the pillars, with a link mechanism comparable to an actual piston-crank mechanism mounted on the upper block. In the link mechanism, a linear guide is used instead of a cylinder, with a block (reciprocating block) having the almost same mass as that of an actual piston block mounted on the rail. The rotating shaft of the DC motor is equipped with a rotating block having an eccentric shaft, which is connected with another shaft mounted on the reciprocating block via a connecting rod. A spherical support element is installed underneath the center of the lower block. The radius of the sphere is designed to position the center of curvature higher than the center of gravity of the drive unit, in order to ensure stable self-standing of the drive unit. This drive unit is designed such that its center of gravity, application point of the unbalance force, and magnitude of the moment of inertia can be changed by adjusting the fixing height of the upper block to the pillars.

On the other hand, the shell is substituted by an aluminum plate, four corners of which are horizontally supported with anti-vibration rubber bushes which are the same as the ones used in an actual compressor. To install the drive unit at the center of this plate, the drive unit and the plate are connected with a resin-made thin plate spring, suppressing the generation of sliding around the contact point and the rotation around the vertical axis.

While driving the link mechanism, the unbalance force caused by the weight of the reciprocating block and the connecting rod is applied on the rotating shaft. This unbalance force generates a friction force at the contact point, which causes vibration transmission. Therefore, in order to counter the unbalance force in the y-axis, we install two balance weights on the rotating block to achieve balance and minimize the friction force in the y-axis acting on the contact point. In order to counter the unbalance force in the x-axis, we design the drive unit to align the application point of the unbalance point with the center of percussion against the contact point to minimize the friction force in the x-axis acting on the contact point.



(a) Simplified model (b) Drive unit

Figure 4: Analytical model

3.2 Drive unit design conditions

This section discusses the design conditions necessary for the drive unit to minimize the natural frequency and the friction force in the *x*-axis acting on the contact point, by using the analytical model shown in Figure 4. In the analysis, as shown in Figure 4 (a), we consider a coordinate system at rest, having the center of sphere O as the origin of the vertical-horizontal plane. We treat both the drive unit and the plate as rigid bodies and move them within the plane. The plate is movable in the *x*-axis only, and the drive unit rolls on the plate without sliding. We eliminate the influence of the plate spring connecting the drive unit and the plate.

With small angular displacement θ of the drive unit, the following equation of motion is obtained.

$$\begin{bmatrix} M+m & -Mh_G \\ -Mh_G & I_G+Mh_G{}^2 \end{bmatrix}\begin{bmatrix} \ddot{x} \\ \ddot{\theta} \end{bmatrix}+\begin{bmatrix} k & 0 \\ 0 & Mg(a-h_G) \end{bmatrix}\begin{bmatrix} x \\ \theta \end{bmatrix}=\begin{bmatrix} -F \\ Fh_P \end{bmatrix}\cos\Omega t \tag{1}$$

The acceleration $\ddot{x}$ and the angular acceleration $\ddot{\theta}$ of the particular solution of equation (1) are applied as follows.

$$\ddot{x}=A_x\cos\Omega t,\quad \ddot{\theta}=A_\theta\cos\Omega t \tag{2}$$

The acceleration amplitude A_x of the plate and the angular acceleration amplitude A_θ of the drive unit are determined from equation (1), as follows.

$$A_x=\frac{Mg(a-h_G)U\Omega^4}{D}+\frac{\{Mh_G(h_P-h_G)-I_G\}U\Omega^6}{D} \tag{3}$$

$$A_\theta=-\frac{kh_PU\Omega^4}{D}+\frac{\{M(h_P-h_G)+mh_P\}U\Omega^6}{D} \tag{4}$$

$$D=\{k-(M+m)\Omega^2\}\{Mg(a-h_G)-(I_G+Mh_G^2)\Omega^2\}-(Mh_G\Omega^2)^2 \tag{5}$$

Here, we focus on the acceleration amplitude A_x of the plate, which is the target for vibration suppression. In the first right member, both the numerator and denominator are of the order of Ω^4. Therefore, after eliminating resonance by reducing the natural frequency to a region outside the rotational speed for operation, reducing the value of $a-h_G$ under the condition of $a>h_G$ can reduce the value of the first right member to within the region of rotational speeds for operation. On the other hand, in the second right member, the numerator is of the order of Ω^6, while the denominator is of the order of Ω^4. Therefore, even if the resonance is eliminated, increasing Ω implies greater amplitude. To avoid this, the following conditional equation is obtained by applying 0 to the coefficient of the numerator of the second right member.

$$Mh_G(h_P - h_G) - I_G = 0 \quad \rightarrow \quad Mh_G(h_P - h_G) = I_G \tag{6}$$

Determining the position of the center of gravity and the moment of inertia of the drive unit that satisfy equation (6) makes the second right member to be 0, regardless of the magnitude of Ω. In the meantime, equation (6) is equivalent to the conditional equation, which aligns the application point P of the unbalance force with the center of percussion against the contact point Q. Aligning the application point P with the center of percussion eliminates the friction force acting at the contact point Q even with the application of the unbalance force, which suppresses the vibration transmission from the drive unit to the plate.

The experimental apparatus shown in Figure 3 can adjust the distance h_P between the contact point and the application point of the unbalance force, by changing the fixing position of the upper block. In the next section, we introduce the design conditions (optimal value) $\hat{h}_P$ with respect to the distance h_P from equation (6), as follows.

$$\hat{h}_P = \frac{I_1 + I_2 + M_1 h_1^2 + M_2 h_2^2}{M_1 h_1 + M_2 h_2} \tag{7}$$

3.3 Results of numerical calculation

To verify the effects of reduction of natural frequency and suppression on the vibration transmission by the self-standing support method, we conducted numerical calculation using the equations introduced in the previous section. Table 1 shows the system parameters of the experimental apparatus. Here, for the sake of discussion, we set the target for the region of low rotational speed of the compressor as 15 Hz.

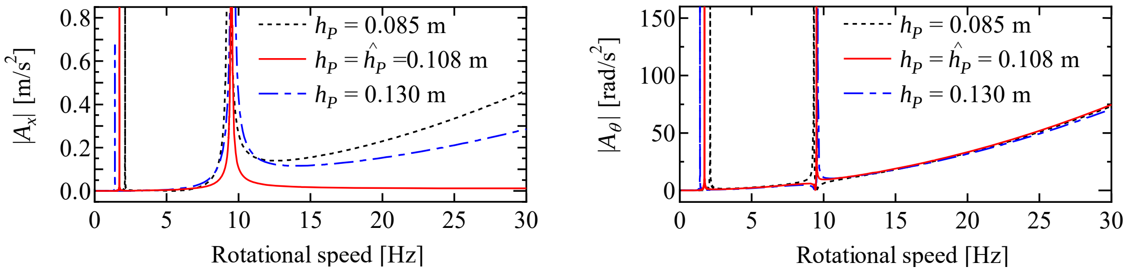
To align the application point of the unbalance force with the center of percussion against the contact point, we obtained $\hat{h}_P = 0.108$ m as the optimal value of distance h_P from equation (7). We then set three values for h_P (0.085 m, 0.108 m, and 0.130 m), including this optimal value, and obtained the acceleration amplitude $|A_x|$ of the plate and the angular acceleration amplitude $|A_\theta|$ of the drive unit from equations (3) and (4) by changing the rotational speed ($\Omega / 2\pi$) of the motor. The resultant response curves are shown in Figures 5 (a) and 5 (b).

Both the response curves $|A_x|$ and $|A_\theta|$ of Figure 5 show the peaks of primary resonance and secondary resonance. As the self-standing support method sufficiently reduces the natural frequency, there is no peak in the region of the targeted rotational speed for operation, namely 15 Hz or above. In the region of 15 Hz or above, when $h_P = 0.085$ m and $h_P = 0.130$ m, both $|A_x|$ and $|A_\theta|$ increased as the rotational speed increased. On the contrary, when $h_P = \hat{h}_P = 0.108$ m, $|A_\theta|$ increased as the rotational speed increased, while $|A_x|$ converged to a smaller value.

As stated above, we confirmed that the acceleration amplitude of the plate $|A_x|$ can be substantially reduced by adopting the center of percussion to the driving unit. In addition, the self-standing support method can simultaneously reduce the natural frequency. By eliminating the natural frequency from the region of rotational speed for operation, the plate vibration can be suppressed for the entire region of rotational speeds for operation.

Table 1: System parameters

M_1 [kg]	2.16	h_1 [m]	0.0291
M_2 [kg]	1.52	h_2 [m]	0.0236
I_1 [kgm^2]	6.59×10^{-3}	a [m]	0.12
I_2 [kgm^2]	1.42×10^{-3}	U [kg· m]	0.4×10^{-3}
m [kg]	2.72	k [N/m]	16,300



(a) Plate (b) Drive unit

Figure 5: Calculated response curves

4. Results of verification experiment

4.1 Vibration reduction effects based on the center of percussion

To verify the validity of the analytical results, we conducted further experiments. As shown in Figure 6, we installed 3D acceleration sensors at the left edge of the lower block and the plate of the drive unit to measure the vibrational acceleration during operation.

We first measured the acceleration signals of the drive unit and the plate during operation, while changing the distance h_P between the contact point and the application point of the unbalance force by adjusting the fixing position of the upper block of the drive unit.

Figure 7 shows the acceleration responses measured at each axis. The results for the drive unit in Figure 7 show that the *x*-axis and *z*-axis acceleration changes in 1.0 m/s^2 or above, while the *y*-axis acceleration stays around 0.2 m/s^2, because the balance weights are appropriately adjusted. The results show that the unbalance force, in association with the reciprocating block motion, is applied to the drive unit in the *x*-axis, creating a rolling vibration in the *x*-*z* planes. In the meantime, the results for the plate in Figure 7 show that the *x*-axis acceleration changes as h_P changes. The dotted line in the figure shows the optimal value of h_P ($\hat{h}_P = 0.108$ m) obtained from equation (7), around the region the *y*-axis acceleration of the plate is about 0.2 m/s^2, which is almost similar to the *y*-axis acceleration of the drive unit. The *z*-axis acceleration of the plate is 0.05 m/s^2 or smaller, suggesting that almost no force acts in the *z*-axis.

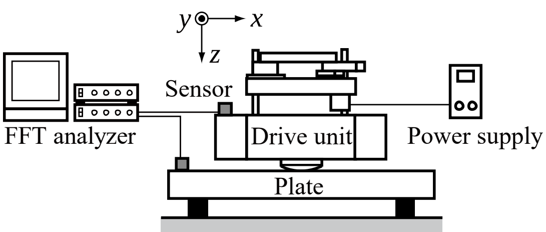


Figure 6: Experimental setup

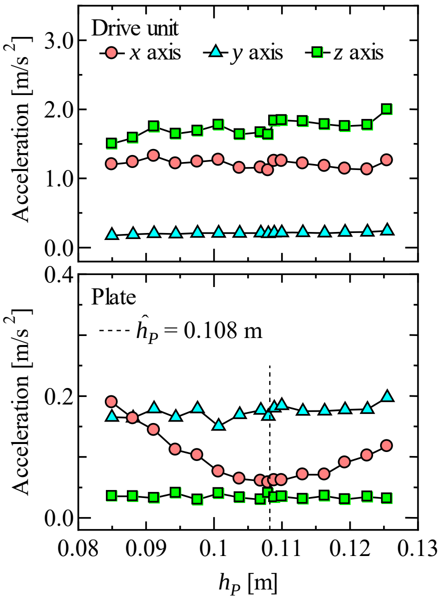


Figure 7: Measured acceleration responses for various values of h_P

Accordingly, it is apparent that by aligning the application point of the unbalance force with the center of percussion against the contact point, the vibration transmission from the driving unit to the plate can be suppressed. In the meantime, the *y*-axis acceleration is slightly greater than the *x*-axis acceleration. In order to reduce this, it is necessary to strictly tune the balance by the balance weight. However, this paper primarily intends to examine the suppression effects on the vibration transmission by adopting the center of percussion. Therefore, in the following discussion, we focus on the *x*-axis acceleration in our verification.

Next, we measured the *x*-axis acceleration signals of the drive unit and the plate with different rotational speeds of a DC motor. To compare with the results of the numerical calculation shown in Figure 5, we set the distance between the contact point and the application point of the unbalance force h_P at two values, namely the optimal value 0.108 m and 0.085 m. Figure 8 shows the obtained results. The horizontal axis in the figure represents the motor rotational speed and the vertical axis represents the axis acceleration of the drive unit and the plate from the top. The dotted line shows the measured natural frequency in the *x*-axis impact test obtained by applying $h_P = \hat{h}_P = 0.108$ m. However, the figure shows only the secondary natural frequency, because the primary and secondary natural frequencies were 1.7 Hz and 9.5 Hz (theoretical values were 1.72 Hz and 9.50 Hz), respectively.

The results for the drive unit show that both the results have similar acceleration responses, that is, increase in acceleration as the rotational speed increases. This indicates that the drive unit has rolling vibration in the *x*-*z* plane. On the other hand, the results for the plate show that the acceleration is 0.1 m/s^2 or less regardless of the rotational speed, if it is set at $h_P = \hat{h}_P = 0.108$ m when the application point of the unbalance force aligned with the center of percussion. However, if it is set at $h_P = 0.085$ m, the acceleration increases as the rotational speed increases. These characteristics are quantitatively similar to the analytical results shown in Figure 5. However, the experimental results do not show the resonance peak at around 9.5 Hz, which is the secondary natural frequency. This reason is supposed to be due to that the unbalance force that excites the force is small in the low rotational speed region and the influence of the damping, which is not considered in Figure 5.

As above, we experimentally verified that the vibration transmission from the drive unit to the plate can be suppressed in the global region of rotational speeds by aligning the application point of the unbalance force with the center of percussion.

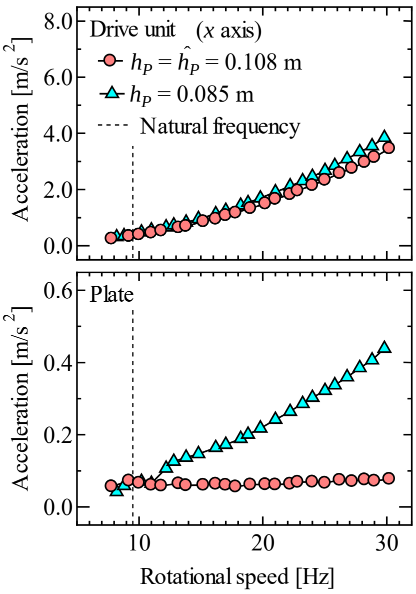


Figure 8: Measured acceleration responses for various values of rotational speed

4.2 Comparison of vibration suppression effects by different supporting methods

We then compared the effects of vibration suppression between the new self-standing support method and the conventional spring support method. In the self-standing support method, the drive unit with distance h_P set at the optimal value $\hat{h}_P = 0.108$ m is used. In the spring support method, four coil springs are installed to support the bottom of the drive unit, similar to that in the self-standing support method.

We first measured the *x*-axis natural frequencies by conducting an impact test on the experimental apparatuses using each support method. The results are shown in Table 2. As described in the previous sections, the natural frequency of the self-standing support (a = 0.12 m) method is reduced to 10 Hz or lower. Furthermore, if the radius of the spherical surface is reduced to a = 0.08 m, the natural frequencies can be further reduced. But, if the value of $a - h_G$ approaches 0 too much, the stability of the self-standing state decrease. On the other hand, the secondary natural frequency of the spring support method is higher than the target value of 15 Hz.

Then, we measured the *x*-axis acceleration of the drive unit and the plate with different rotational speeds of a DC motor. Figure 9 shows the obtained results. The horizontal axis in the figure represents the motor rotational speed and the vertical axis represents the axis acceleration of the drive unit and the plate from the top. The dotted line shows the natural frequency of the spring support method.

The results for the drive unit show that both the support methods produced similar acceleration responses. On the other hand, the results for the plate show that the acceleration with spring support method is greater in the region of 25 Hz or lower. In particular, the acceleration increases up to around 0.3 m/s^2 due to resonance at around the natural frequency. During the operation of actual compressors at low rotational speeds, significant vibration is produced at

Table 2: Comparison of natural frequencies

Order	Self-standing support [Hz]		Spring support [Hz]
	a =0.12 m	a = 0.08 m	
1st	1.7	1.0	8.4
2nd	9.5	8.6	16.9

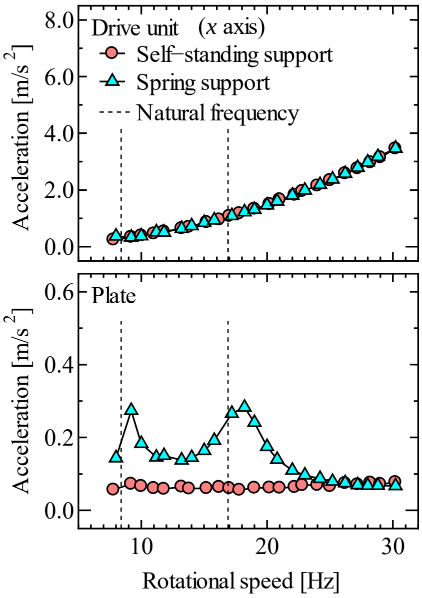


Figure 9: Comparison between the acceleration responses

around the natural frequency. To avoid this, the natural frequency must be eliminated from the region of rotational speeds for operation.

As described in the previous sections, in the self-standing support method, the natural frequency is reduced to the target value of 15 Hz or lower, creating no resonance. In addition, by adopting the center of percussion, the vibration of the plate is suppressed to 0.1 m/s^2 or lower, even if the vibration of the drive unit increases as the rotational speed increases.

As described above, we verified that the self-standing support method has better vibration suppression effects on the plate compared with the spring support method.

5. CONCLUSIONS

To reduce the vibration of reciprocating compressors, we proposed a self-standing support method that supports the drive unit in a compressor shell. We verified its validity with respect to vibration suppression effects through experiments as well as numerical calculations using a simplified model that was created based on an actual compressor structure consisting of the drive unit and the plate. The results are as follows.

(1) The self-standing support method attaches a spherical support element to the bottom of the drive unit, which is directly placed on the plate. The method does not require conventional support springs, and thus it can substantially reduce the natural frequency. Therefore, the entire natural frequency can be eliminated from the region of rotational speeds for operation. We verified that the creation of resonance-caused vibration was avoided in the region of low rotational speeds.

(2) Aligning the application point of the unbalance force with the center of percussion against the contact point of the sphere eliminates the friction force acting at the contact point even with the generation of the unbalance force, suppressing the vibration transmission from the drive unit to the plate. Accordingly, we verified that the vibration transmission to the plate could be sufficiently suppressed, even if the unbalance force acting at the drive unit increases in the region of high rotational speeds.

NOMENCLATURE

M	Mass of Drive unit	(kg)
m	Mass of plate	(kg)
k	Spring constant	(N/m)
I	Moment of inertia	(kgm^2)
h	Height	(m)
a	Radius of Spherical surface	(m)
x	Horizontal displacement of plate	(rad)
θ	Angular displacement of drive unit	(rad)
F	Unbalance force	(N)
Ω	Angular velocity of the motor	(rad/s)
U	Amount of unbalance	(kgm)

Subscript

1	Lower unit
2	Upper unit
G	Center of gravity of driving unit

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