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A STUDY OF A LINEAR COMPRESSOR WITH A GAS SPRING

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ABSTRACT

In conventional linear compressors, a mechanical resonant spring, such as a leaf spring or a coil spring, is used to oscillate a piston. The problem with mechanical resonant springs is that high reliability cannot be easily achieved for large piston stroke, and they cannot easily be made more compact. On the other hand, a gas spring has the possibility to solve these problems. However, ordinary gas spring is not suited for a long life operation due to gas leakage. Therefore, to apply a gas spring to a resonant spring of a linear compressor, a self-compensation mechanism for gas leakage is necessary. In this report, a prototype of a linear compressor with a gas spring provided with a self-compensation mechanism was fabricated based on a theoretical analysis and its performance was evaluated. The results were that the prototype had the potential to achieve the performance equivalent to that of a conventional linear compressor with a mechanical spring.

NOMENCLATURE

m	mass of moving part	l_c	length of compression chamber seal
$x(t)$	position of moving part	l_g	length of gas spring chamber seal
B	magnetic flux density	$P_c(t)$	pressure in compression chamber
l	length of motor coil	$P_g(t)$	pressure in gas spring chamber
$i(t)$	electric current	P_s	suction pressure
μ	oil viscosity	R	motor resistance
δ	clearance	L	motor inductance
d_c	piston diameter	$V(t)$	voltage
d_g	plunger diameter		

1. INTRODUCTION

Due to increasing interest in energy saving and environmental protection, there is demand for the development of higher efficiency compressors for air conditioners and refrigerators. Linear compressors are paid attention from the viewpoint of efficiency, since a piston is driven directly by a linear motor without a crank mechanism^[1]. In a linear compressor, a resonant spring is used to obtain a piston stroke with small thrust of a linear motor. Normally, leaf springs or coil springs are used as resonant springs^[1,2]. However, their main drawback is that when the spring constant is raised to increase resonant frequency, allowable bending of the spring is reduced, and the spring constant and the allowable bending tend to be mutually exclusive. In other words, to increase refrigerant circulation, both the resonant frequency and piston stroke must be increased, but when one of these is increased, the other must be decreased. For this reason, there is interest in the development of a resonant spring compatible with these conditions, particularly in linear compressors for air conditioners, which have a larger refrigeration capacity than refrigerators.

Then, the authors tried to use a gas spring as a resonant spring of the linear compressor. A gas spring can be designed at any spring constant simply by changing the diameter of the plunger, and can be made compact and to have a large spring constant without compromising the stroke. In addition, the gas spring is very cost effective, because plunger of the gas spring can be integrally formed with the piston of the compression chamber, and other components are very simple shape.

In this paper, a prototype of a linear compressor with a gas spring based on a theoretical analysis was fabricated, and its feasibility was examined.

2. STRUCTURE OF THE COMPRESSOR

Figure 1 shows the structure of the linear compressor. In the Figure, (a) is a mechanical spring type and (b) is a gas spring type. A mechanical spring type uses leaf springs or coil springs ^[1,2]. In a gas spring type, a gas spring consisted of a plunger and a cylinder, is provided opposite the compression chamber.

In a gas spring, enclosed gas leaks through the clearance between the plunger and cylinder, and the spring constant gradually changes. In the gas spring type linear compressor, this problem is solved by employing a self-compensating design to replace any gas leakage by using a refrigerant gas as the enclosed gas and providing a simple valve mechanism, consisting of a reed valve covering a hole on the end of the gas spring cylinder. If any enclosed gas leaks out from the gas spring chamber, the reed valve opens when the spring chamber gas falls below suction pressure, drawing in more refrigerant gas. This mechanism ensures that the gas spring maintains a constant quantity of gas.

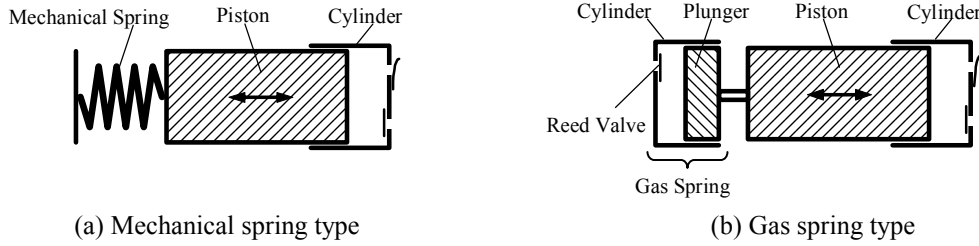


Figure 1: Structure of the linear compressor

3. THEORETICAL ANALYSIS

Analysis Method

Prior to fabricating the prototype, to decide the design parameters and to confirm its operation theoretically, the dynamic analysis of structural parts and the leakage flow analysis of gas leakage from the clearance were carried out. Figure 2 shows the dynamic model of the compressor. In the dynamic model, every force applied to the moving part, including the force due to the differential pressure applied to the piston and the plunger, oil viscosity force, motor thrust, etc. is considered and an equation of motion (1) is described using the dynamic model. For the pressure in the compression chamber $P_c(t)$, the pressure change at each stage of compression, discharge, re-expansion, and suction calculated from the moving part position $x(t)$ was applied so as to describe the nonlinear gas spring effect of the compression chamber. Similarly, for the pressure in the gas spring chamber $P_g(t)$, the pressure change at each stage of compression, expansion, and suction calculated from the moving part position $x(t)$ was applied. Equation (1) was solved by Runge-Kutta-Gill method.

$$m \frac{d^2 x(t)}{dt^2} = Bli(t) - \pi(d_c l_c + d_g l_g) \mu \frac{dx(t)}{dt} - \pi \left(\frac{d_c}{2} \right)^2 \{P_c(t) - P_s\} + \pi \left(\frac{d_g}{2} \right)^2 \{P_g(t) - P_s\} \quad (1)$$

Equation (2) shows a voltage equation of equivalent circuit to the motor part ^[3].

$$Bl \frac{dx(t)}{dt} + Ri(t) + L \frac{di(t)}{dt} = V(t) \quad (2)$$

The flow rate of the refrigerant leakage between the piston and cylinder in the compression chamber and between the plunger and cylinder in the gas spring was calculated in leakage flow analysis, using Ishii et al.'s method ^[4], in which leaked refrigerant from the clearance is treated as an incompressible and viscous fluid. In addition, pressure in the compression chamber $P_c(t)$ and the gas chamber $P_g(t)$ were compensated by taking the refrigerant leakage into account.

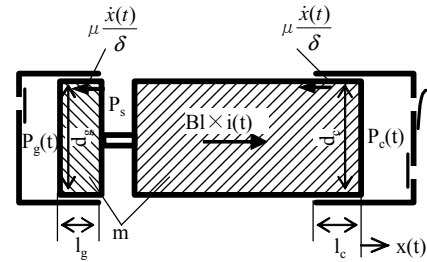


Figure 2: Dynamic model of the compressor

Results of Analysis

The compressor is assumed to be for a room air conditioner using the refrigerant R410A. The operational conditions are shown in Table 1. The specifications of the compressor used for the theoretical analysis are shown in Table 2. The diameter of the piston was set at 35 mm, equal to that of a mechanical spring type linear compressor^[5] experimentally produced in advance. The calculation was performed to find the optimum values of the diameter of the gas spring and the moving distance of the piston and plunger, in the ranges of 30-60 mm and 20-60 mm, respectively. Here, the moving distance is the range wherein the moving part can move.

Table 1: Operational conditions (R410A)

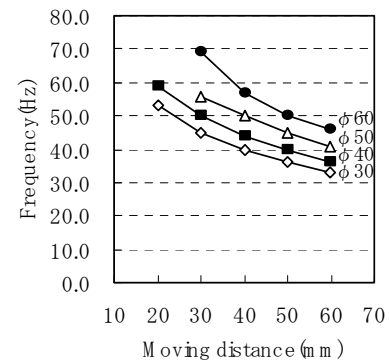
Suction pressure	1.21 MPa (abs)
Discharge pressure	2.58 MPa (abs)
Suction temp.	24.0 °C
Exp. inlet temp.	32.9 °C
Refrigerating capacity	2.5kW

Table 2: Specifications of the compressor

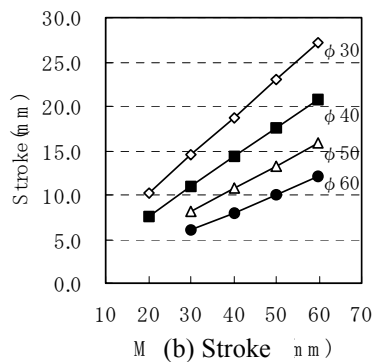
Piston diameter	ϕ 35mm
Gas spring diameter	ϕ 30,40,50,60mm
Moving distance	20,30,40,50,60mm
Piston clearance	15 μ m
Gas spring clearance	30 μ m
Mass of moving part	2.2kg
Motor thrust constant	56N/mm

Figure 3 shows the influence of the gas spring diameter and the moving distance on the behavior of the compressor with respect to (a) resonant frequency, (b) stroke, (c) top clearance of the compression chamber, and (d) maximum pressure of the gas spring chamber.

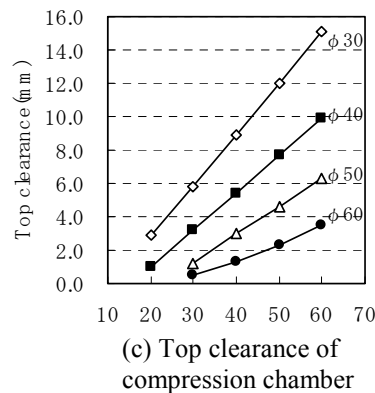
From Figs 3(a) through 3(d), it can be seen that when the gas spring diameter increases, the resonant frequency also increases, the stroke and top clearance decrease, and the maximum pressure in the gas spring chamber also decreases. When the moving distance increases, the maximum pressure in the gas spring chamber changes very little, while the resonant frequency decreases and the stroke and top clearance increase. The resonant frequency, stroke, clearance, and the maximum pressure relate to each other.



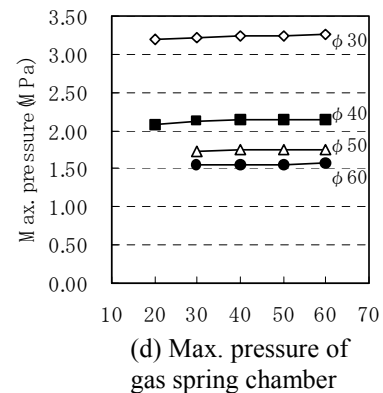
(a) Resonant frequency



M (b) Stroke mm)



(c) Top clearance of compression chamber



(d) Max. pressure of gas spring chamber

Figure 3: Behavior of the compressor vs. moving distance and gas spring diameter

Concerning the influence of the gas spring diameter, the larger the gas spring diameter, the larger the reaction force for the same stroke. When the reaction force increases, the resonant frequency also increases as the spring constant increases. In addition, the larger the gas spring diameter compared with the piston diameter, the smaller the top clearance, since the moving part oscillates in a closer position to the compression chamber due to the differential pressure. The smaller the top clearance, the higher the volumetric efficiency of the compression chamber and the shorter the stroke. The shorter the stroke, the lower the maximum pressure in the gas spring chamber.

As for the influence of the moving distance, the larger the distance, the larger the top clearance of the compression chamber and gas spring chamber. When the top clearance of the compression chamber increases, the stroke also increases as the volumetric efficiency decreases. For a gas spring, however, since the stroke and top clearance increase simultaneously, both the compression ratio and the maximum pressure in the gas spring chamber are almost constant. In addition, if the gas spring stroke increases, the spring constant decreases because the maximum pressure does not change, and the resonant frequency decreases.

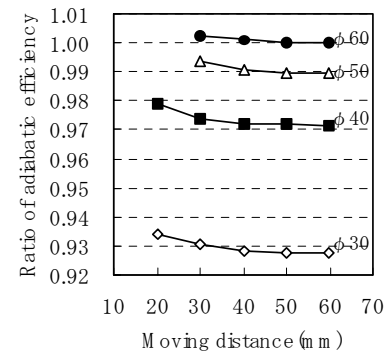
Figure 4 shows the influence of gas spring diameter and the moving distance on (a) adiabatic efficiency, (b) mechanical efficiency, (c) motor efficiency, (d) compressor efficiency. Enclosed gas leakage from the gas spring chamber is taken into account in the adiabatic efficiency calculations, and only copper loss is taken into account in the motor efficiency. Each efficiency is shown as the ratio to the efficiency where the gas spring diameter is 60 mm and the moving distance is 60 mm.

As illustrated in Fig. 4(a), the adiabatic efficiency depends mainly on the gas spring diameter, the larger the diameter, the higher the efficiency. This is thought to be because, as shown in Fig. 3(d), the larger the gas spring diameter, the lower the maximum pressure and the less the enclosed gas leakage due to the differential pressure.

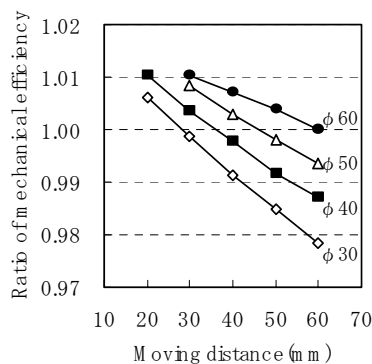
As shown in Fig. 4(b), the mechanical efficiency increases with decreased the moving distance and increased gas spring diameter. This occurs because the stroke decreases in these cases, shown in Fig. 3(b).

As seen in Fig. 4(c), the shorter the moving distance and the larger the gas spring diameter, the lower the motor efficiency. This is because although the stroke decreases in these cases, for the same motor output, the shorter the stroke, the larger the motor thrust required, leading to increased electric current and copper loss, shown in Fig. 3(b).

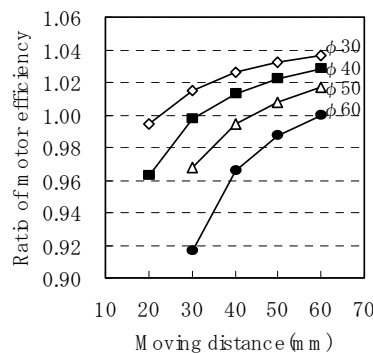
As shown in Fig. 4(d), the compressor efficiency in the calculation range, which is the product of adiabatic efficiency, mechanical efficiency, and motor efficiency, reaches a maximum when the gas spring diameter is 60 mm and the moving distance is 60 mm. If the moving distance is larger than 60 mm, the compressor efficiency will be higher, but the gas spring becomes impracticably large. Consequently, the gas spring and the moving distance are both set at 60 mm.



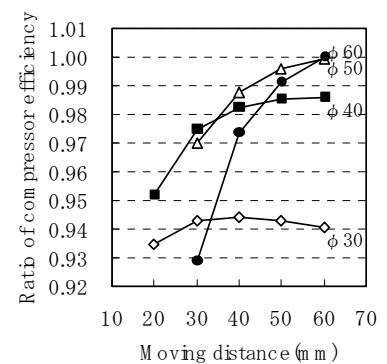
(a) Ratio of adiabatic efficiency



(b) Ratio of mechanical efficiency



(c) Ratio of motor efficiency



(d) Ratio of compressor efficiency

Figure 4: Efficiency vs. moving distance and gas spring diameter

Figure 5 shows the frequency characteristics of the compressor efficiency, refrigeration capacity, and input when the gas spring diameter is 60 mm, the moving distance is 60 mm, and the motor thrust is constant. Each value is the ratio to the maximum. The refrigeration capacity and the input are the largest at 54 Hz and 55 Hz, respectively. Although the compressor efficiency is the largest at 49 Hz, change around this frequency is less dramatic than that seen in the refrigeration capacity and input.

The above-mentioned examination reveals that there is a frequency at which each parameter of refrigeration capacity, input, and compressor efficiency reach a peak. Hence resonance also occurs in the gas spring type linear compressor.

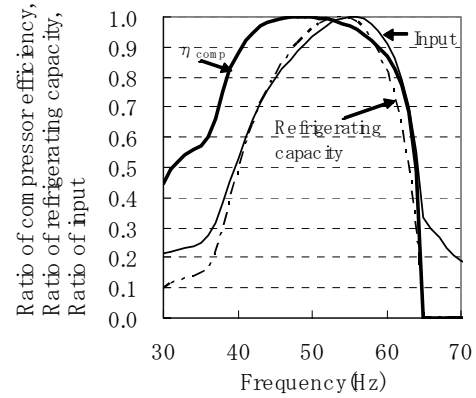


Figure 5: Ratio of compressor efficiency, refrigerating capacity, and input vs. frequency

4. PROTOTYPE

Figures 6 and 7 show the cross-section and appearance of the gas spring prototype, respectively. The prototype specifications are shown in Table 3.

The prototype was driven by AC single-phase inverter, and refrigeration capacity was obtained by controlling the amplitude and frequency of the sinusoidal wave.

A pressure sensor and a position sensor were built into the prototype to measure the pressure in the compression chamber and gas spring chamber and the position of the moving part.

Table 3: Specifications of the

Compressor type	Hermetically sealed type
Motor	Moving magnet type
Refrigerating capacity	0.5~3.0kW
Piston diameter	φ 35mm
Gas spring diameter	φ 60mm
Moving distance	60mm

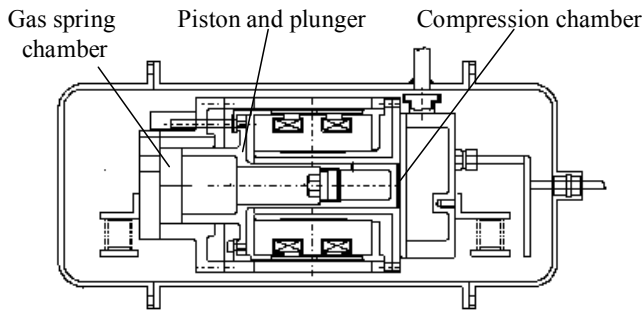


Figure 6: Cross-section of the prototype

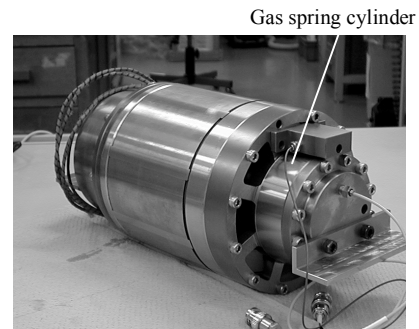


Figure 7: Appearance of the prototype (without pressure vessel)

5. EXPERIMENTAL EVALUATION

Trial operation

The prototype was operated under the conditions shown in Table 1. However, the performance could not be evaluated because the moving part vibrated in a position closer to the gas spring chamber than predicted in the theoretical analysis and the top clearance of the compression chamber became very large.

As the reason why the moving part vibrated at a different point from that predicted theoretically, it is thought that the theoretical analysis was not accurate enough with respect to oil viscosity and quantity of refrigerant gas leakage from the compression chamber and gas spring chamber. The accuracy will be improved by use of feedback from the experimental results of the prototype.

In order to evaluate the performance of the prototype, a force generated by adding a DC component to the AC current was given to the moving part to offset the vibration point toward the compression chamber. Table 4 shows a comparison of the experimental results between the gas spring prototype with added DC and a conventional mechanical spring type compressor. The efficiency of the gas spring prototype was 84% of that of a conventional model.

Table 4: Experimental

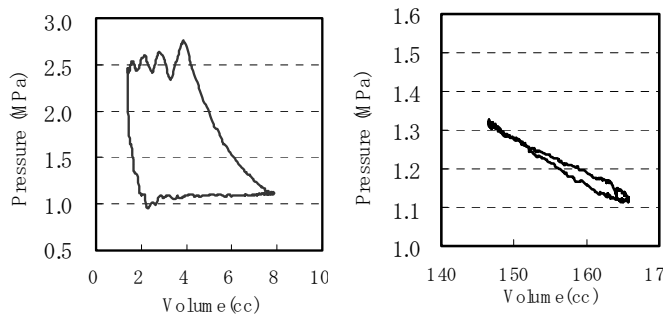
		Mechanical spring type	Gas spring type
Ratio of compressor efficiency		100%	84%
Frequency		79Hz	57Hz
Electric current	AC	8.4A	11.0A
	DC	-	3.1A

Loss Analysis

Figure 8(a) shows a P-V diagram for the compression chamber. The compression power obtained from the P-V diagram was 61.8% of total input. Of this, theoretical compression power was 53.4% and adiabatic loss was 8.4% of total input.

Figure 8(b) shows a P-V diagram of the gas spring chamber. A loss of 3.3% of total input was caused in the gas spring prototype by enclosed gas leakage. The spring constant was approximately 89 N/mm. The copper loss of the motor calculated from the current waveform including the resistance and AC/DC components of the coil was 9.9% of total input. The inverter loss, calculated from the measurements at the entrance and exit of the inverter, was 9.6% of total input. The mechanical loss and the iron loss of the motor obtained as residual loss was 15.3% of total input.

The results of the loss analysis are shown in Fig. 9. The friction loss between the plunger and cylinder in the gas spring was not analyzed, since it is included in the mechanical loss. However, since the plunger of the gas spring is integrally formed with the piston of the compression chamber, and the piston and cylinder with small clearance pivotally supports the integrated piston/plunger, the friction loss of the gas spring is estimated to be smaller than that between a piston and cylinder.



(a) Compression chamber

(b) Gas spring chamber

Figure 8: P-V diagram

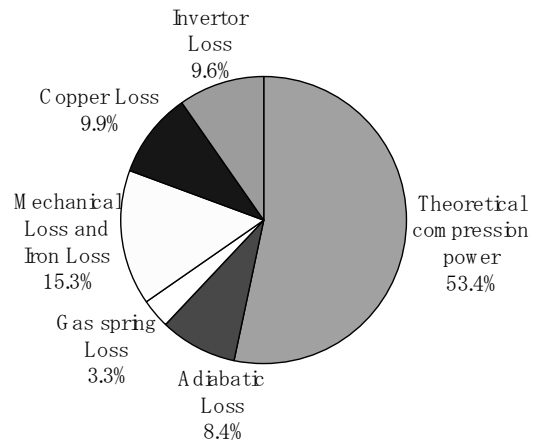


Figure 9: Loss analysis

Comparison with Mechanical Spring Type

Fig. 10 shows a comparison of the inverter loss, the copper loss of the motor, and other losses between the gas spring prototype and conventional mechanical spring type compressor under the operational conditions listed in Table 1. As shown in the Figure, the prototype has a larger inverter loss and copper loss of the motor. This increase is considered to be due mainly to the DC component used to offset the moving part. By further optimization of the gas spring diameter and the moving distance, this DC component can be eliminated. Therefore, the gas spring type linear compressor is considered to have the potential to achieve the performance equivalent to that of the mechanical spring type, and to be very suitable for linear compressors used in air conditioners from the viewpoint of high reliability for large stroke and large spring constant.

In addition, from the results of loss analysis described in the preceding section, it can be understood that loss of the gas spring was caused by the enclosed gas leakage, which accounts for 3.3% of total input, and friction loss between plunger and cylinder. The total of these losses is considered to be larger than the loss caused by mechanical spring hysteresis. Nevertheless, the total loss other than inverter loss and motor copper loss in the gas spring type compressor is equivalent to that of the mechanical spring type. The reason is thought to be that although the side force of the mechanical spring generated by bending increases the friction loss between the piston and cylinder, this side force of the gas spring is small since the gas spring is completely symmetrical about its axis.

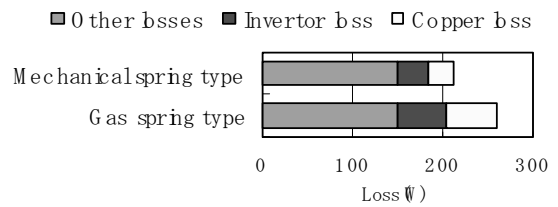


Figure 10: Comparison of losses

6. CONCLUSIONS

A prototype of a linear compressor with a gas spring, which can secure high reliability even in the case of a large stroke and large spring constant, and can be very cost effective, was fabricated based on a theoretical analysis and its performance was evaluated. The results are as follows:

- It was confirmed both theoretically and experimentally that a gas spring can be used as a resonant spring of a linear compressor.
- The prototype revealed that loss due to enclosed gas leakage of the gas spring is as little as 3.3% of the total input.
- Optimizing the design parameters, the gas spring type linear compressor has the potential to achieve the performance equivalent to that of the mechanical spring type, and is very suitable for a linear compressor used in air conditioners from the viewpoint of reliability.

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