Performance Characteristics of a Refrigerator-Freezer with Parallel Evaporators using a Linear Compressor

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ABSTRACT

A linear compressor for a domestic refrigerator-freezer has energy saving potential compared with a reciprocating compressor because of a low friction loss and free piston system. A linear compressor can control the piston stroke since it does not have mechanical restriction of piston movement. Therefore, the energy consumption of a domestic refrigerator-freezer using a linear compressor can be reduced by changing the cooling capacity of the compressor. In order to investigate the performance of a refrigerator-freezer with parallel evaporators using a linear compressor and the relation between cooling capacity of the linear compressor and cooling load, experimental simulation is conducted with variation of the capacity of a linear compressor, an ambient temperature, and cooling load. In addition, the power consumption of a linear compressor is compared to that of an inverter reciprocating compressor in a refrigerator-freezer. The performance of a linear compressor is measured with variation of the capacity of a linear compressor from 60% to 100% of the maximum capacity in a refrigerator-freezer. Based on the experimental data, the power consumption of a linear compressor is reduced by 22.4% with 70% capacity compared to 100% but on-time ratio is increased by 12.8%.

1. INTRODUCTION

Recently, a linear compressor has been adopted for domestic refrigerator-freezers due to its potential to reduce the energy consumption with some refrigerator manufacturers. However, in many refrigerator manufacturers, a reciprocating compressor is used for a domestic refrigerator-freezer. The mechanical loss of conventional reciprocating compressors is higher than that of a linear compressor because of four friction regions to generate friction loss by a crank-driven mechanism. On the contrary, a linear compressor has one friction region between a piston and a cylinder. Therefore, a liner compressor is the most efficient compressor due to its low friction loss, simple refrigerant flow path and highly efficient linear motor (Kim and Jeing, 2013). In addition, the noise level of linear compressors is lower than that of conventional reciprocating compressors due to the small number of friction regions. The linear compressor does not have mechanical restriction to piston movement. Hence, the cooling capacity of a linear compressor can be modulated by a piston position controller in accordance with the variation of

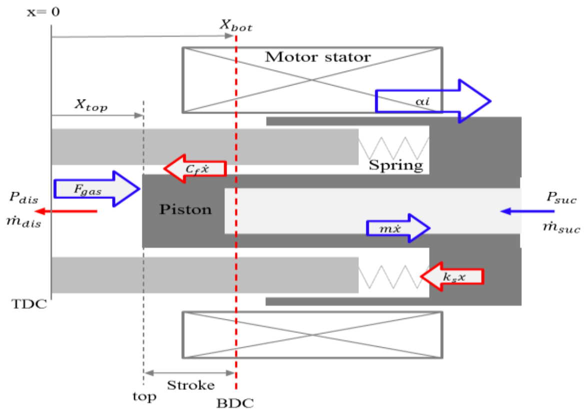

Figure 1 Schematic diagram of a linear compressor

cooling demand. Lee *et al*. (2000) applied Triac, which control the AC voltage in order to control piston stroke. Chun and Ahn (2008) applied a PWM(Pulse Width Modulation) inverter to the compressor. However, stroke controllers such as Triac and PWM inverter need costs to be applied to the compressors and power to run. The control logic is complicated because the piston stroke is influenced by many operating conditions. Kim *et al.* (2011) investigated an inherent capacity modulated(ICM) linear compressor without using stroke controllers. This compressor without a drive has a great advantage to reduce power consumption and cost and enhance compatibility with various types of refrigerators. In this paper, the possibility of self-modulation for cooling capacity is presented. The stroke increases with reduction of the gap between TDC and the piston. The smaller this distance is, the greater capacity the compressor can produce. The linear compressor has several advantages such as potential to reduce energy consumption and self-modulation. However, there is hardly any study on the performance characteristics of a refrigerator-freezer using a linear compressor in the literature.

The objective of this work is to investigate the performance of the refrigerator-freezer with using a linear compressor. The operating characteristics of the linear compressor are figured out with variation of the ambient temperature and compressor cooling capacity in a refrigerator-freezer with parallel evaporators. In addition, the power consumption of a linear compressor in a refrigerator-freezer is compared to that of an inverter reciprocating compressor in order to find advantages of the linear compressor.

2. RELATED THEORIES FOR A LINEAR COMPRESSOR

Fig.1 shows a simple schematic diagram of a linear compressor. A linear compressor consists of a piston, a mechanical spring, a suction and a discharge valve and a linear motor. The piston is driven by the linear motor and moved from the top dead center(TDC) to bottom dead volume(BDC). An oscillating motion is generated by the moving magnet of the linear motor in linear fashion. The stroke of the piston can change due to no mechanical restriction of movement.

The discharge pressure increases with an increase of ambient temperature because pressure of discharge and condenser are directly influenced by the ambient temperature. The gas force and the gas spring acting on the piston are increased with rising discharge pressure. The gas spring under rising discharge pressure increases with an increase of the ambient temperature. The average gas spring, $K_{gas,ave}$ is expressed by Eq.(1).

$$K_{gas,ave} = \frac{\sum (P_n - P_{n-1}) A_p / (x_n - x_{n-1})}{\sum N_i} = \frac{P_{dis} - P_{suc}}{X_{bot} - X_{top}} \qquad (1)$$

where $K_{gas,ave}$ is the average value of the gas spring, P_{dis} is the discharge pressure, and P_{suc} is suction pressure. Eq.(2) can be formulated with combination of the mechanical, electrical and thermodynamic system.

$$I = \frac{X\left(\left(K_m+K_{gas}-m\omega^2\right)^2+\left(\left(C_f+C_{gas}\right)\omega\right)^2\right)}{\alpha\sqrt{\left(K_m+K_{gas}-m\omega^2\right)^2+\left(\left(C_f+C_{gas}\right)\omega\right)^2}} \tag{2}$$

where X is the stroke amplitude, α is a motor parameter and C_f is the friction damping coefficient. The current increases as the gas spring rises like expressed in Eq.(2)

The stroke amplitude is decided by the design of the electrical impedance with a given voltage and frequency like Eq.(3) and Eq.(4).

$$X = \int\left(V_{in} - Ri - Ie^{j\omega t}\left(L\omega - \frac{1}{C\omega}\right)\right) \tag{3}$$

$$imp_{lc} = \left(L\omega - \frac{1}{C\omega}\right) \tag{4}$$

When imp_{lc} is less than 0, the stroke of the piston increases with increasing current. The details of the ICM linear compressor can be found in a literature (Kim *et al*. 2011). As described above, the linear compressor has an ability of self-modulation with variation of cooling demand.

3. EXPERIMENTAL SET-UP AND TEST PROCEDURE

3.1. Experimental set-up

In this experiment, parallel evaporator cycle is adopted with internal volume of $0.87m^3$ $(870L)$. Figure 2 shows schematic diagram of the experimental apparatus. The experimental apparatus consist of a linear compressor, a condenser, two evaporators, two capillary tubes and SLHX(Suction Line Heat Exchanger) with a 3-way valve and a check valve. The specifications of the experimental set-up are shown in Table 1. In parallel evaporator cycle, R- and F-cycles operate independently for each compartment cooling load with a compressor. Therefore, a 3-way valve should be installed at inlet of R- and F-capillary tube to separate refrigerant flow path into R- and F-evaporators. A check valve is also installed at outlet of F-evaporator to prevent back flow into F-evaporator during off-time because pressure in a F-evaporator is lower than a R-evaporator. The compressor capacity is controlled with a compressor driver from full capacity of the linear compressor to 60% of the full capacity, which change the piston stroke. A condenser is installed in a duct in order to assess an effect of the flow rate on the condensing temperature. The flow rate for a condenser is controlled with a blow fan and an orifice flow meter. Hot-line is connected between the outlet of condenser and the inlet of capillary tube. This is attached inside front surface of the refrigerator case in order to defrost between door gasket and refrigerator case and obtain sub-cooling degree in a refrigerator. The refrigerant mass flow rate is measured with a coriolis mass flow meter(Oval engineering, CA001, Accuracy $\pm$0.2%). The mass flow meter is installed at outlet of hot-line because the accuracy of the mass flow meter is higher at liquid state of the operating fluid. K-type Thermocouples and pressure transducers are installed at inlet and outlet of each component. The pressure transducers have an accuracy of $\pm$1.0%. In addition, the experimental apparatus is installed in a constant temperature and humidity room to maintain constant ambient temperature and relative humidity.

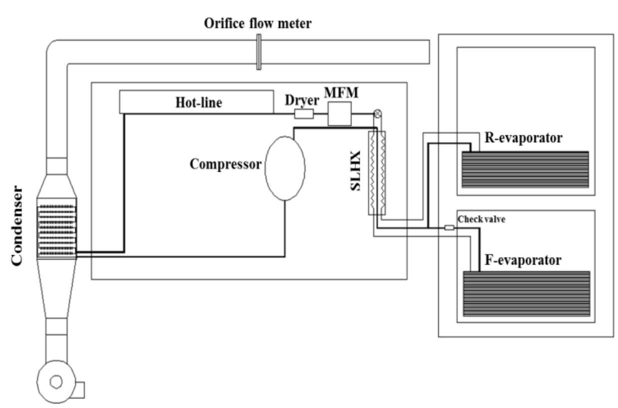

Figure 2 Schematic diagram of experimental apparatus

Table1 Specifications of each component

	Specifications			
Compressor	Linear compressor,(Displace volume : 16.5cc)			
Condenser	Spiral 10(R)	Width (mm)	Depth (mm)	Height (mm)
		200	180	180
Evaporator	F-room	707	60	180
	3(R) 6(C)			
	R-room	736	90	120
	3(R) 4(C)			
Capillary tube	Freezer	Capillary Tube 0.75mm(D)		
	Refrigerator	Capillary Tube 0.85mm(D)		

3.2. Test procedure

The performance of the refrigerator cycle with a linear compressor is measured by varying an ambient temperature, cooling capacity of the linear compressor, and cooling load in a R-compartment. The ambient temperature changes from 15˚C to 30˚C with relative humidity of 60%. The variation of the ambient temperature affects the cooling load for the each compartment. In this test, total power of a compressor and on-time ratio are measured and integrated for four hours after the variation of temperatures at every measured point and input power of a linear compressor is almost constant. In addition, refrigerant mass flow rate and piston stroke of a linear compressor are measured by a coriolis mass flow meter and driver software for a linear compressor. During this test, setting temperatures of F-compartment and R-compartment are -18 $\pm$0.5˚C and 3$\pm$1.5˚C, respectively. The temperatures of each compartment for a cyclic operation are measured at center of compartments. In addition, the compressor frequency is constant, which yields 100% of the full capacity but the stroke of a piston changes owing to increasing condensing pressure.

The performance of the refrigerator is measured with variation of compressor cooling capacity from 60% to full capacity at an ambient temperature of 25˚C and a relative humidity of 60% (ISO 15502, N-class condition).. In this test, cooling time for the F-compartment temperature to reach setting temperature from the ambient temperature is measured. The heat of 100W is suddenly put in R-compartment by a heater under cyclic steady condition. This heat is internal cooling load as door opening and putting foods. In cooling load test, compressor power consumption and time for the R-compartment temperature to reach setting temperature of 3$\pm$1.5˚C are measured in the R/F simultaneous operation.

In the parallel cycle, refrigerant is trapped in a F-evaporator because pressure in a F-evaporator in lower than that in a R-evaporator. Hence, refrigerant recovery operation should be carried out after F-operation. In refrigerant recovery operation, the 3way-valve closes all refrigerant flow paths and a compressor operates simultaneously.
The optimum charge amount of the refrigerant is selected based on the test result to yield lowest power consumption. Figure 3 shows the power consumption and cooling time to reach the setting temperature in a F-compartment with

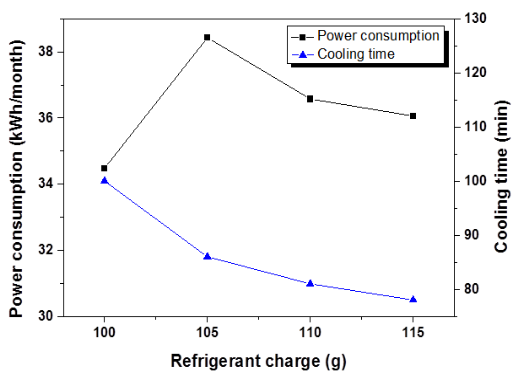

Figure 3 Power consumption and cooling time with refrigerant charge

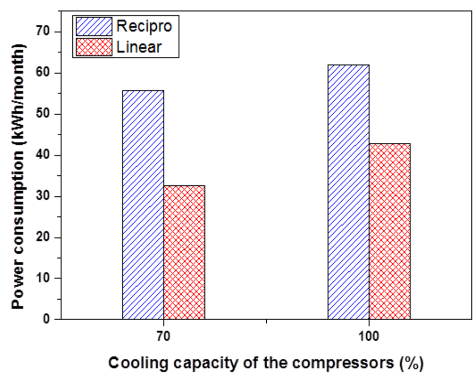

Figure 4 Power consumption with compressor capacity

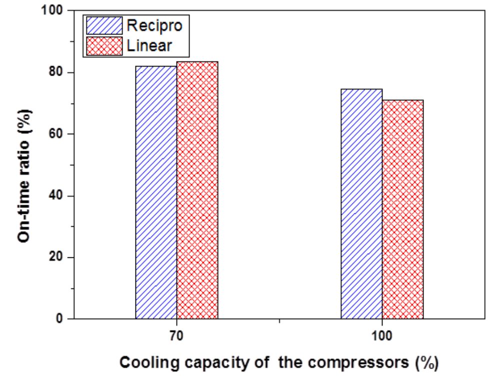

Figure 5 On-time ratio with compressor capacity

varying charge amount. As shown in Fig.3, the lowest power consumption is observed at the refrigerant charge of 100g but the cooling time to reach setting temperature of the F-compartment is relatively longer than other cases of 105g, 110g and 115g. When the charge amount of the refrigerant is over 120g, a suction temperature of the compressor decreases sharply below 0℃ in F-operation mode. If a suction temperature is too low compared to ambient temperature, the compressors may be damaged. Hence, the refrigerant charge of 115g is selected as the optimum amount for the experimental apparatus.

4. RESULTS AND DISCUSSION

4.1 Reciprocating compressor vs linear compressor

The performance of a linear compressor is compared to that of a reciprocating compressor with variation of compressor capacity in the ambient temperature of 25˚C and relative humidity of 60%. Figure 4 and 5 show power consumption and on-time ratio for two compressors with change of compressor cooling capacity. The power consumption of a linear compressor is lower than that of a reciprocating compressor with 41.7% and 31.1% for two cases although on-time ratio is almost same in two compressor tests.

4.2 Effects of the ambient temperature

The refrigerator performance is compared with variation of the ambient temperature with using a linear compressor. Figure 6 shows power consumption of a linear compressor in the refrigerator. Power consumption increases linearly

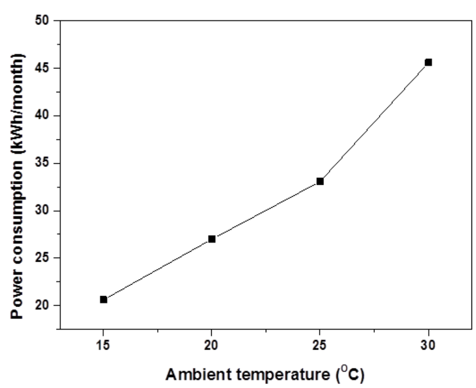

Figure 6 Power consumption with ambient temperature

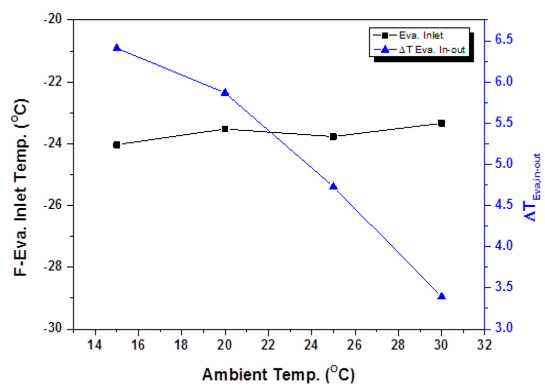

Figure 7 $T_{evp,in}$ and $\Delta T_{evp,in-out}$

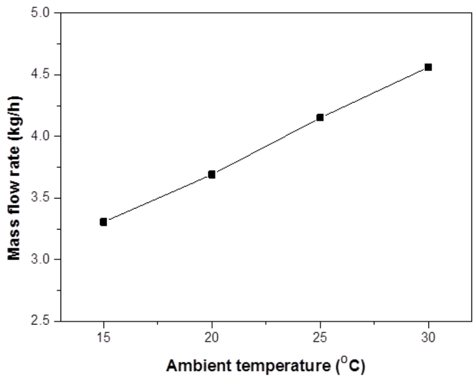

Figure 8 Refrigerant mass flow rate

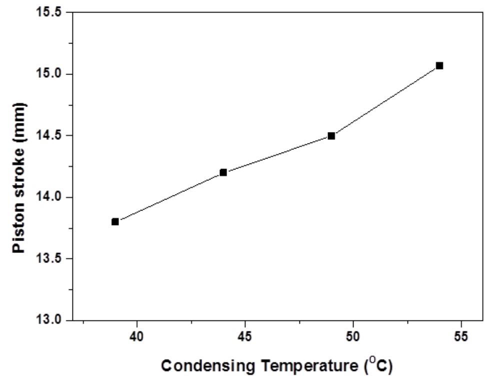

Figure 9 Piston stroke with condensing temperature

with increasing the ambient temperature from 15˚C to 25˚C. However, the power consumption increases sharply at 30˚C of ambient temperature. This is because the saturated pressure of R600a is not linear function with the saturated temperature. Hence, the ratio of suction pressure to discharge pressure increases with increasing condensing temperature, causing high compression work. Figure 7 shows variation of a temperature at F-evaporator inlet and temperature difference between inlet and outlet of a F-evaporator during only F-operation. As shown in Figure 7, temperature difference of F-evaporator decreases with increasing the ambient temperature while inlet temperature of F-evaporator maintains almost constant temperature. Figure 8 shows the refrigerant mass flow rate measured at outlet of hot-line. As shown in Figure 8, the refrigerant mass flow rate increases with an increase of ambient temperature. This is because piston stroke increases according to an increase of ambient temperature. As described in chapter 2, piston stroke increases with a rise of ambient temperature. Figure 9 shows variation of piston stroke with change of condensing temperature in calorimeter tests for a linear compressor. The piston stroke increases with an increase of condensing temperatures, which means an increase of the cooling capacity of the compressor.

4.3 Effects of cooling capacity of a linear compressor.

The cooling capacity of a linear compressor can change with variation of a piston stroke. The compressor should be operated with consideration for cooling loads such as ambient temperature, putting the foods and door opening.

In this test, cooling time and power consumption of a linear compressor are measured with variation of cooling capacity of the compressor at ambient temperature of 25˚C and relative humidity of 60%. As shown in Figure 10, cooling times to reach setting temperature of -18˚C in freezer compartment with compressor cooling capacity of 100, 90, 80, 70, 60% are 73, 91, 105, 115 and 138min, respectively. Figure 11 shows the power consumption and on-time

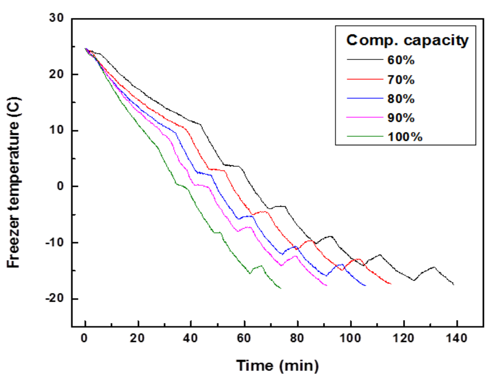

Figure 10 Temperature in F-compartment

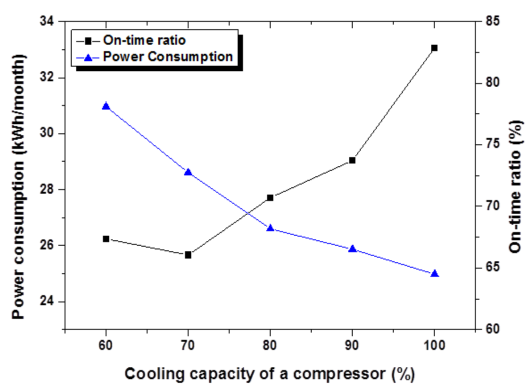

Figure 11 Power consumption and on-time ratio

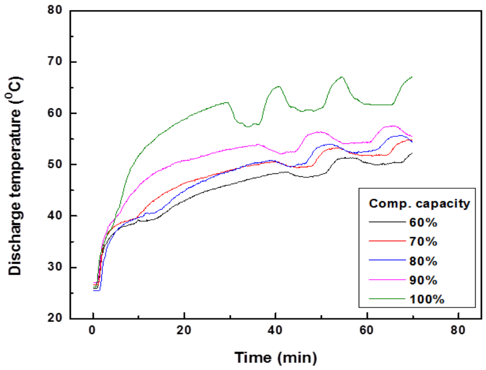

Figure 12 Discharge temperatures with compressor capacity

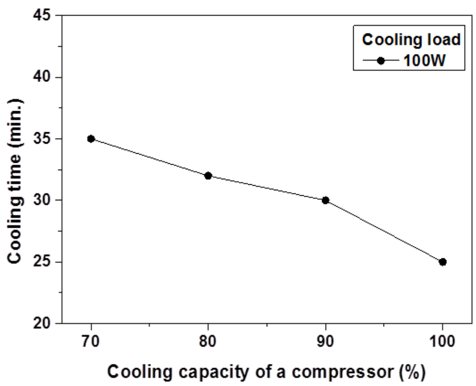

Figure 13 Cooling time with cooling load

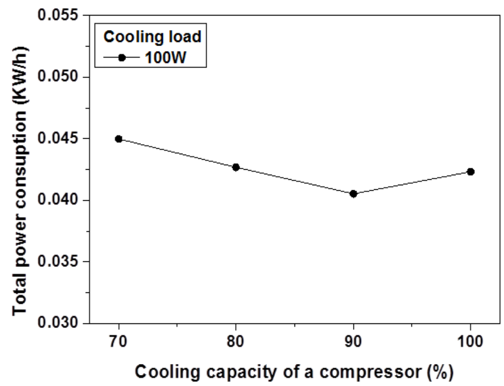

Figure 14 Total power consumption with cooling load

ratio of a linear compressor. On-time ratio decreases with an increase of compressor cooling capacity and power consumption increase from 70% to 100%. However, power consumption at 60% cooling capacity is larger than that at 70% cooling capacity because cooling capacity of a linear compressor is not sufficient compare to the cooling load. Also, the power consumption of a compressor is sharply increased at 100% cooling capacity compared to 90%. Figure 12 shows discharge temperatures of a linear compressor with variation of compressor cooling capacity. The discharge temperature is sharply increased at 100% cooling capacity due to increasing compression work and friction loss and low efficiency of a compressor at full capacity. As shown in Figure 11, the power consumption of a linear compressor was reduced by 22.4% with 70% capacity compared to 100% but on-time ratio was increased by 12.8%. As shown in Figure 11, power consumption and on-time ratio are in trade-off relationship. Therefore, the refrigerator must be designed with consideration of this relationship between power consumption and on-time ratio.
In the tests on effects of cooling capacity of a linear compressor, internal cooling load of 100W for 12min. is inputted by a heater installed in a R-compartment under the steady operating condition. Figure 13 and 14 show cooling time and total power consumption to remove the cooling load. As shown in Figure 13 and 14, total power consumption decreases with increase of compressor cooling capacity from 70% to 90%. However, the total power consumption at 100% increases while cooling time decreases.

5. CONCLUSIONS

The experimental simulation of a refrigerator-freezer using a linear compressor is conducted with R600a. As results of the experimental simulation, the power consumption of a linear compressor is 41.7% and 31.1% lower than that of an inverter reciprocating compressor for cooling capacity of 70% and 100% of full capacity of both compressors.

The cooling capacity of a linear compressor increases with an increase of ambient temperature owing to an increase of piston stroke. Also, the power consumption can be reduced by controlling cooling capacity of a linear compressor. The power consumption and on-time ratio are in trade-off relationship. Therefore, a linear compressor should be operated with consideration for this relationship.

NUMENCLATURE

A_p	area of piston	(m^2)
C_f	friction damping coefficient	(Nm/s)
C_g	gas damping coefficient	(Nm/s)
i	current	(A)
$K_{gas,ave}$	average gas spring	(N/m)
k_{gas}	gas spring constant	(N/m)
k_m	resonant spring constant	(N/m)
L	inductance	(mH)
P_n	nth pressure in cylinder	(Pa)
P_{dis}	discharge pressure	(Pa)
P_{suc}	suction pressure	(Pa)
R	resistance	(Ω)
V	input voltage	(V)
x_{bot}	piston's bottom position	(m)
x_o	initial displacement	(m)
x_{top}	piston's top position	(m)
α	motor constant	(N/A)
ω	angular velocity	(rad/s)

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