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ABSTRACT

A comprehensive model of a linear compressor for electronics cooling was previously presented by Bradshaw et al. (2011). The current work utilizes this model to explore the energy recovery characteristics of a linear compressor as compared to those of a reciprocating compressor. An analysis of the impact of dead (clearance) volume on both a linear and reciprocating compressor is presented. It is shown that the volumetric efficiency of linear and reciprocating compressors displays similar trends as the amount of dead volume in the cylinder is increased. However, the overall performance of the linear compressor remains relatively unaffected by an increase in dead volume while a reciprocating compressor suffers from a proportional decrease in overall performance. It is shown that this is due to the ability of a linear compressor to recapture energy of the compressed gas during the expansion process. This peculiar behavior allows a linear compressor to be used for efficient capacity control. An improved linear compressor design is proposed for an electronics cooling application, with a predicted cooling capacity of 200 W, a cylindrical compressor package size of diameter 50.3 mm, and a length of 102 mm.

1. INTRODUCTION

A comprehensive simulation model for a miniature-scale linear compressor was recently developed by Bradshaw et al. (2011). The model was also validated against experiments conducted on a prototype linear compressor constructed for the purpose. Additionally, recent work by Bradshaw et al. (2012) presented an analysis which utilized this comprehensive model to study the sensitivity of the linear compressor to changes in various geometric parameters. The sensitivity studies conducted showed that the linear compressor is highly sensitive to changes in the leakage gap between the piston and cylinder as well as the spring eccentricity; both parameters should be minimized for optimal performance. Therefore, it is important to quantify and control these parameters in any compressor design that is mass-produced to maximize performance. Additionally, these studies showed that it is advantageous to keep the stroke-to-diameter ratio relatively small. The tradeoff to this small stroke-to-diameter ratio is an increase in leakage due to the increased leakage area. Therefore, a tradeoff exists between overall performance and leakage which will depend strongly on the leakage gap between the piston and cylinder.

The previous study also described confounding results caused by an increase in the clearance volume (dead volume) between the piston and valve plate. The results showed that the overall efficiency of a linear compressor does not seem to be heavily influenced by an increase in this parameter. This suggests that a linear compressor could be utilized effectively for capacity control. However, the previous study did not verify this hypothesis in detail.

The sensitivity analysis presented here examines the performance of a linear compressor with variable compressor dead volume to simulate a variable stroke compressor. These results are used to simulate a variable-capacity miniature-scale refrigeration system for an electronics cooling application. In addition, these results are compared with a reciprocating compressor for comparison to a traditional technology. Finally, this knowledge is combined into a revised design of a linear compressor for an electronics cooling application.

2. DEAD (CLEARANCE) VOLUME/STROKE CONTROL

To simulate a variation in compressor capacity, the compressor model is operated with varying amounts of dead (clearance) volume, which is depicted for a linear compressor in Figure 1. This is used to simulate a compressor with a variable stroke. An actual linear compressor with a fixed cylinder size would change capacity by reducing the compressor stroke. However, since varying the stroke will also change the calculated resonant frequency, it becomes difficult to directly compare the compressor performance metrics from one stroke input to the next, as they depend on the stroke and operating frequency (Bradshaw et al. 2012). Thus, an increasing compressor dead volume is used to represent a decreasing compressor stroke. The same piston diameter and stroke are used as in the spring eccentricity and leakage gap studies presented in Bradshaw et al. (2012), of 1.24 cm and 2.54 cm, respectively. This leads to a fixed displaced/swept volume for the compressor. The fixed values of spring eccentricity, motor efficiency, and leakage gap are 0.5 cm, 0.9, and 3 μm, respectively. The dead volume in the compressor is varied from 10% of displaced volume to 120% of displaced volume. In addition, to explore any influence of friction, the dry friction coefficient **f** was varied between 0.1 and 0.3. The design condition utilized is the same as presented in Bradshaw et al. (2012): 20 °C, 40 °C, and 5 °C for evaporating, condensing, and superheat temperatures, respectively.

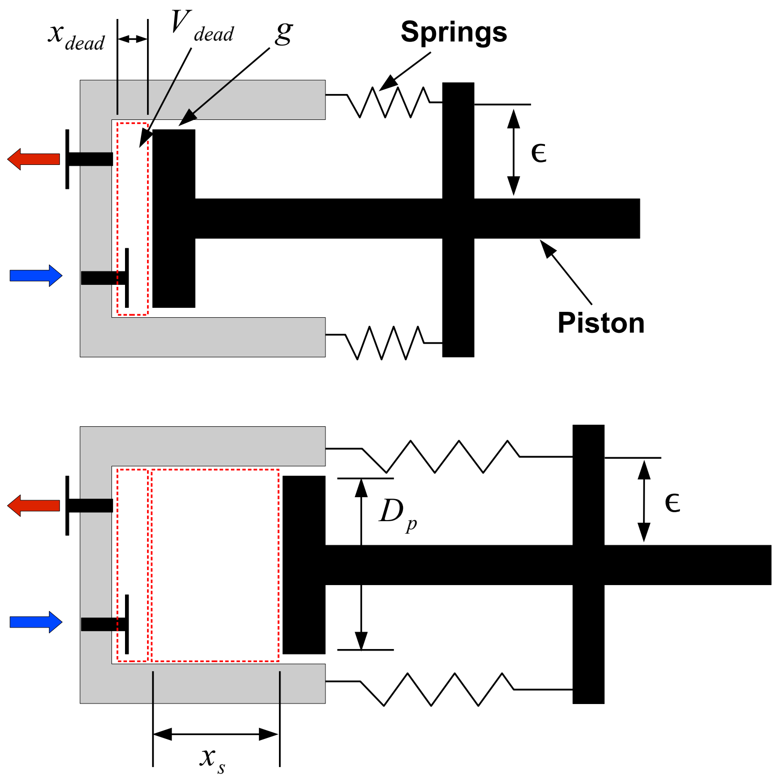

Figure 1: Schematic diagram of linear compressor at Top Dead Center (TDC, top) and Bottom Dead Center (BDC, bottom) with primary linear compressor components and design parameters highlighted.

Figure 2 shows the volumetric efficiency as a function of compressor dead volume. As the dead volume increases the volumetric efficiency decreases linearly. With 0.3 cm^3 of dead volume the compressor operates at approximately 90% volumetric efficiency, while at 3.6 cm^3 dead volume the compressor operates at approximately 5% volumetric efficiency. The three separate dry friction coefficients cannot be distinguished in Figure 2 as the volumetric efficiency is very weakly dependent on friction.

The overall isentropic efficiency is also explored as shown in Figure 2. At a dead volume of 0.3 cm^3 at each dry friction coefficient the efficiency is maximized. The overall isentropic efficiency decreases slightly until a dead volume of roughly 2.5 cm^3, beyond which the overall isentropic efficiency degrades rapidly. It is also noted that as the dry friction coefficient decreases, the overall performance increases.

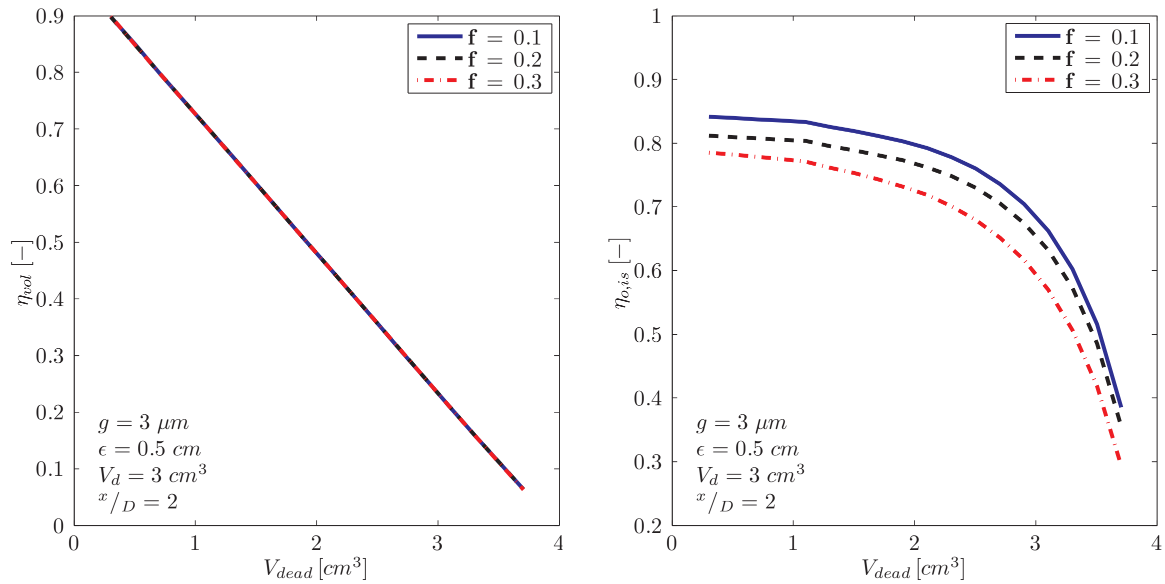

Figure 2: Volumetric (left) and overall isentropic (right) efficiencies as function of compressor dead volume for three dry friction coefficients.

Results from this study show that a linear compressor behaves differently from a typical positive displacement compressor. The trend in the volumetric efficiency is a result of a decay in the amount of massflow provided by the compressor as the dead volume increases. The cause of this relates to the change in volume required to compress a fixed mass of gas. This can be seen by examining an idealized compression process, modeled by a polytropic process and introducing an expression for total volume which includes the dead volume:

$$V_r = \left(\frac{P_2}{P_1}\right)^{1/n} + \frac{V_{dead}}{\pi r_p^2 x_{TDC}}\left[\left(\frac{P_2}{P_1}\right)^{1/n} - 1\right]. \tag{1}$$

where V_r is the volume ratio, the ratio of maximum to minimum compression volume:

$$V_r = \frac{A_p x_{BDC}}{A_p x_{TDC}} = \frac{x_{BDC}}{x_{TDC}}. \tag{2}$$

Equation (1) shows that as the dead volume increases from zero (dead volume cannot be less than zero) that the volume ratio required to obtain a certain pressure rise increases proportionally. Therefore, for a fixed volume and pressure ratio, as the dead volume increases the massflow obtained will decrease proportionally.

The overall isentropic efficiency would be expected to follow a similar trend. However, as seen in Figure 2 the trend is not linear. As the dead volume increases from 0.3 cm^3 to about 2.5 cm^3 the overall efficiency shows only minor degradation. Figure 2 shows that the massflow, on the other hand, is drastically reduced with the same change in dead volume. It may be concluded that the compressor produces less massflow rate at the higher dead volume but also requires less power. This behavior can then be exploited if the system level performance is considered, as discussed in the following section.

3. CAPACITY CONTROL

A reduction in massflow translates into a reduction in refrigeration capacity. By analyzing the refrigeration system for electronics cooling shown in Figure 3 the system cooling capacity may be calculated as follows:

$$\dot{Q}_{cool} = \dot{m}(h_1 - h_4). \tag{3}$$

If the system operates under fixed environmental conditions the enthalpies entering and exiting the system evaporator (h_4 and h_1) will remain constant, where h_4 is calculated assuming an isenthalpic throttling process:

$$h_4 = h_3 = h(T = T_{sat} - \Delta T_{sub}, P_2) \tag{4}$$

and ΔT_{sub} is assumed to be 10 °C. This means that the system cooling capacity is proportional to the massflow generated by the compressor. The power required to drive the compressor is calculated using the comprehensive linear compressor model, and the system Coefficient of Performance (COP) is defined as:

$$COP = \frac{\dot{Q}_{cool}}{\dot{W}_{in}}. \tag{5}$$

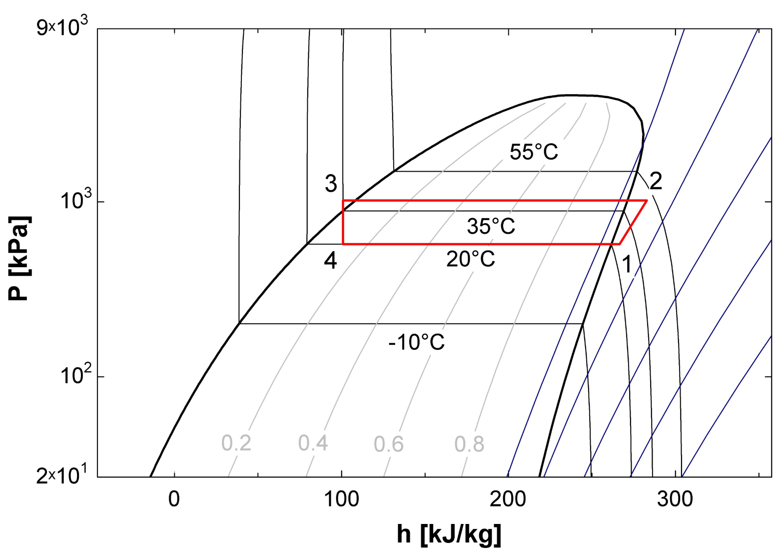

Figure 3: Pressure enthalpy diagram of typical R134a miniature-scale refrigeration cycle for electronics cooling.

For relative comparison to a reversible refrigeration cycle the COP of a Carnot refrigeration cycle is utilized:

$$COP_{carnot} = \frac{T_{evap}}{T_{cond} - T_{evap}}. \tag{6}$$

Using the Carnot COP the second law effectiveness of the refrigeration cycle is defined as follows:

$$\varepsilon_{carnot} = \frac{COP}{COP_{carnot}}. \tag{7}$$

Using the results from Section 2 as an input with a dry friction coefficient of 0.1, a variable capacity system may be simulated, with the variable massflow achieved by increasing the compressor dead volume. Figure 4 shows the predicted system COP and second law effectiveness ε as a function of the cooling capacity.

At the smallest cooling capacity the linear compressor produces the smallest massflow rate, which translates to the largest dead volumes in Figure 2. As the cooling capacity increases, the dead volume decreases and subsequently the massflow increases. In addition, the overall efficiency of the compressor also increases as the dead volume decreases which generates a better system COP. Figure 4 shows that an electronics cooling system can provide a reasonably high system COP over a wide range of cooling capacities. This simulated compressor operates relatively well for cooling capacities from roughly 200 W to over 600 W, which would provide a sufficient amount of variation to maintain consistent cooling for an electronic system, based on the predictions from the International Roadmap of Semiconductors (2011 ed.). This is another useful feature of the linear compressor for an electronics cooling application.

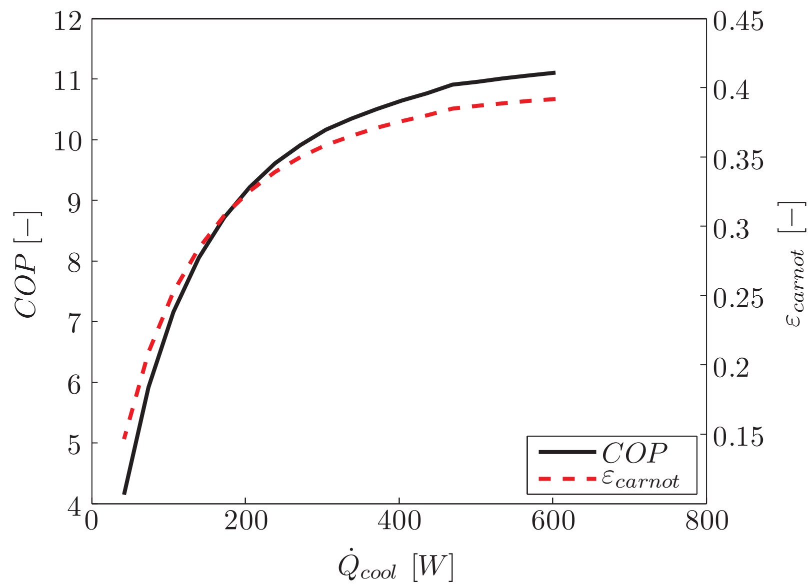

Figure 4: Coefficient of Performance and second law effectiveness of a R134a refrigeration cycle operating at a typical electronics cooling condition with 10 °C condenser subcooling and variable capacity predicted by the linear compressor model.

4. Comparison of a Linear Compressor to Reciprocating Compressor Behavior

The unique behavior of a linear compressor shown in the previous sections serves as motivation to compare a linear compressor to a more well-known compressor technology such as the reciprocating compressor. A reciprocating compressor provides a good benchmark for comparison to a linear compressor since the compression mechanism of the two devices may be readily compared. Fundamentally, linear and reciprocating compressors are very similar as they are both piston-cylinder devices. However, due to the kinematics of a reciprocating compressor the displaced volume is dictated by the design of the compressor and is not a function of the operating conditions or power input. Figure 5 shows a kinematic diagram of a reciprocating compressor driven with torque T at rotational speed ω.

Assuming that the compressor components act as rigid bodies, an expression for the motion of the piston as a function of the crank angle may be written as:

$$x_p = L_1 \cos(\beta_1) + \sqrt{L_2^2 - L_1^2 \sin^2(\beta_1)}. \tag{8}$$

Differentiating this expression twice, an expression for the acceleration of the piston is obtained:

$$\ddot{x}_p = -L_1\omega^2 \cos(\beta_1) - \frac{L_1^2\omega^2 \cos(2\beta_1)}{\sqrt{L_2^2 - L_1^2 \sin^2(\beta_1)}} - \frac{L_1^4\omega^2 \cos(2\beta_1)}{2\sqrt[3]{L_2^2 - L_1^2 \sin^2(\beta_1)}}. \tag{9}$$

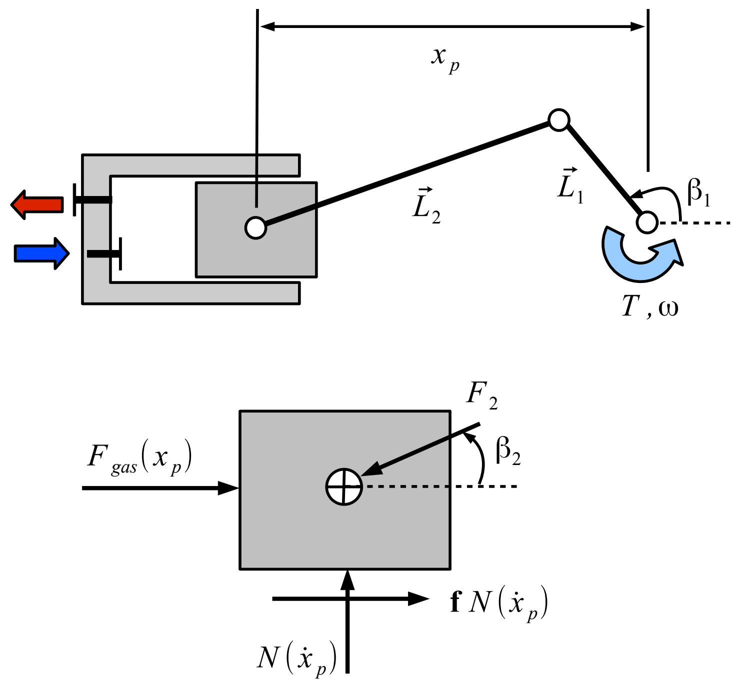

Figure 5: Schematic diagram of reciprocating compressor mechanism and free body diagram of compressor piston assuming dry friction contact between piston and cylinder wall.

Additionally, an expression for the equation of motion of a reciprocating compressor may be constructed using the free body diagram given in Figure 5:

$$\underbrace{M_{mov}\ddot{x}_p}_{\text{Intertia}} + \underbrace{\mathbf{f}F_2 \sin(\beta_2)}_{\text{Damping}} + \underbrace{k_{gas}x_p}_{\text{Stiffness}} = F_2 \cos(\beta_2) \tag{10}$$

and noting that

$$k_{gas}\mathrm{x}_p = \mathrm{P}_{cv}\mathrm{A}_P. \tag{11}$$

This equation shows that a reciprocating compressor displays a similar equation of motion compared with the piston of a linear compressor, given by Bradshaw et al. (2011) as

$$\underbrace{M_{mov}\ddot{x}_p}_{\text{Inertia}} + \underbrace{c_{eff}\dot{x}_p}_{\text{Damping}} + \underbrace{(k_{gas} + k_{mech})x_p}_{\text{Stiffness}} = k_{mech}\epsilon\theta + F_{drive} \tag{12}$$

$$J_{CG}\ddot{\theta} + k_{mech}\epsilon^2\theta = k_{mech}\epsilon x_p \tag{13}$$

Equations (12) and (13) show the two-degree-of-freedom equations of motion for a linear compressor piston and highlight the inertial, damping, and stiffness terms. The reciprocating compressor also has inertial, damping, and stiffness terms associated with it. However, unlike a linear compressor a reciprocating compressor has a stiffness that is only associated with the gas compression and is thus much smaller than for a linear compressor (*i.e.* $k_{linear} >> k_{recip}$). This small stiffness term makes it very difficult to operate a reciprocating compressor at a resonant frequency as it would require operation at a slower speed than A/C rotary motor technology typically operates (30 - 60 Hz). In addition, the leakage in a compressor is adversely impacted by the result of slowing operational speed which is another reason why operating at lower speeds is typically avoided. Equation (10) is further expanded to determine the required torque from the crank. Realizing that β_2 can be written in terms of β_1 as

$$\tan\beta_2 = \frac{\left(L_1 / L_2 \right)\sin\beta_1}{\sqrt{1-\left[\left(L_1 / L_2 \right)\sin\beta_1\right]^2}}, \tag{14}$$

the torque required for piston movement may be calculated by solving the equation of motion described in Equation (10) for F_2 and using geometry to solve for the torque at the crank:

$$\mathbf{T} = (1+\mu)\left[(M_{mov}\ddot{x}_p + k_{gas}x_p)x_p \tan\beta_2\right]. \tag{15}$$

The power required to drive the reciprocating compressor is then given as:

$$\dot{W}_{in,recip} = T\omega. \tag{16}$$

The overall isentropic efficiency of the reciprocating compressor may then be expressed as:

$$\eta_{o,is,recip} = \frac{\dot{m}(h_{2,s} - h_1)}{\dot{W}_{in,recip}}. \tag{17}$$

Using these calculations a direct comparison between a reciprocating and a linear compressor may be made. This is accomplished by simulating a compressor and calculating the work input using Equation (16) for the reciprocating compressor and using the methods described by Bradshaw et al. (2011) for a linear compressor. With respect to leakage, a linear compressor is similar in behavior to a reciprocating compressor, and is not considered in this comparison. The frictional losses are also not considered here and the same friction coefficient, **f** = 0.1, is used for analysis of the reciprocating compressor and linear compressor. The sensitivity studies presented in Bradshaw et al. (2012) are not reproduced here with reference to reciprocating compressor performance. However, changes with respect to dead volume, or capacity control, are explored as this represents one of the significant differences between a reciprocating compressor and a linear compressor.

The variation in dead volume for the reciprocating compressor analysis matches that of a linear compressor. It is observed that the volumetric efficiency follows the same trend for both compressors. Therefore, the trends predicted in Figure 2 are the same for both compressors as each tends to lose mass flow as dead volume increases for the reasons explained in Section 2. However, when comparing the overall isentropic efficiency, Figure 6 shows that as the dead volume increases the overall isentropic efficiency of a reciprocating compressor tends to decrease linearly, in a similar fashion as the volumetric efficiency.

The reason for the difference is noted in the equations of motion for the two compressors shown in Equations (12) and (10), where the stiffness term in the linear compressor is much larger due to the mechanical springs. The stiffness in a mechanical system is analogous to a capacitance in an electrical system, and provides a mechanism for a mechanical system to store energy. Therefore, the linear compressor has a higher ability to store energy, or mechanical capacitance. This proves useful when the compressor operates with a variable stroke (or with variable dead volume) as the linear compressor can recapture energy imparted to the gas. The trend of the overall isentropic efficiency of a reciprocating compressor is a result of the power required to drive the compressor mechanism remaining relatively constant, but the net massflow delivered decreasing, as dead volume increases. Conversely, the linear compressor still shows the same reduction in mass flow as the reciprocating compressor. However, as a result of the high amount of mechanical capacitance, it is able to recapture some of the power typically lost during gas re-expansion and thus operate at a lower power input at smaller capacity levels (*i.e.*, partial load).

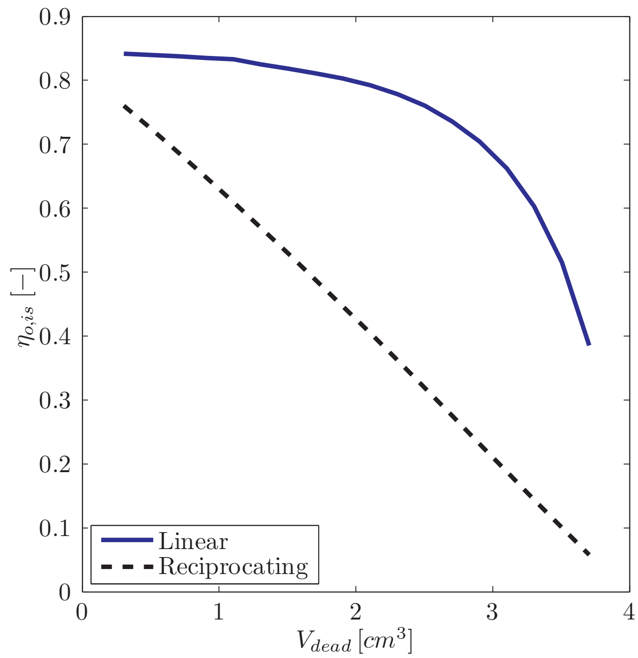

Figure 6: Comparison of overall isentropic efficiency as a function of compressor dead volume for a linear compressor and a reciprocating compressor.

5. AN IMPROVED LINEAR COMPRESSOR DESIGN FOR ELECTRONICS COOLING

Based on the results obtained in this work as well as in Bradshaw et al. (2012), an improved linear compressor design is formulated for an electronics cooling application. A compressor for such an application should be compact, as package size in electronic equipment is often a constraint. The performance of the previous prototype design by Bradshaw et al. (2011) may be readily improved but with the total compressor package size being decreased. The modified design presented here provides an improved overall efficiency and reduced package size.

Similar to the design presented in Bradshaw et al. (2011), the improved linear compressor design utilizes an off-the-shelf motor from HW2 Technologies. A cooling capacity of 200 W is selected to represent a desktop computer cooling application. The design condition utilized is the same as in the sensitivity studies: 20 ^{o}C, 40 ^{o}C, and 5 ^{o}C for evaporating, condensing, and superheat temperatures, respectively. The comprehensive compressor model is then run using the design parameters listed in Table 1 with various stroke-to-diameter ratios necessary to achieve the required 200 W of cooling. The required stroke-to-diameter ratio and the corresponding continuous driving force dictate the choice of an appropriate H2W Technologies motor (P/N NCM02-17-035-2F). The modeling results from the present work show an increase in compressor performance with a decrease in the stroke-to-diameter ratio. However, the force required to operate the compressor piston increases with a decrease in this ratio. An increase in force requires a larger linear motor, which translates into a larger overall package size. Thus, there is a trade-off between package size and performance. For the current design, a stroke-to-diameter ratio of 0.4 leads to an acceptable overall package size.

Table 1: Key design parameters of the improved linear compressor design compared with the design presented in Bradshaw et al. (2011).

	k_{mech}	**f**	D_p	g	f_{res}	ϵ
Design	N/mm	-	cm	μm	Hz	cm
Current Work	30.6	0.2	1.35	4	60	0.5
Bradshaw et al. (2011)	23.0	0.35	1.217	13	44.5	1.145

The proposed design of the improved miniature linear compressor is shown in Figure 7. The overall performance of the compressor as well as the corresponding refrigeration system performance are listed in Table 2. The improved design illustrates the capability of the technology to be miniaturized without compromising overall performance.

Table 2: Predicted performance of the improved linear compressor design compared with the design from Bradshaw et al. (2011).

	x/D	$\dot{Q}_{cool}$	η_{vol}	$\eta_{o,is}$
Design	-		-	-
Current Work	0.4	200	0.96	0.86
Bradshaw et al. (2011)	2	520	0.4	0.08

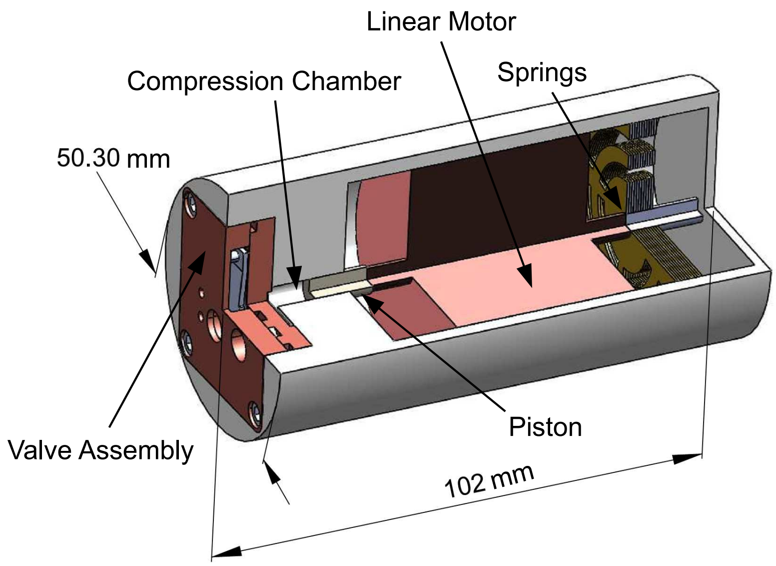

Figure 7: Improved linear compressor design with predicted cooling capacity of 200 W (units in mm).

6. CONCLUSIONS

This work demonstrates the ability of the linear compressor to recapture energy typically lost during the expansion process. This ability gives the linear compressor the unique ability to operate efficiently over a wide range of dead (clearance) volumes. This ability may be utilized for capacity control in a miniature-scale refrigeration system for electronics cooling, which necessitates a certain degree of capacity control. This work also illustrates the ability of the linear compressor to be readily scaled to smaller capacities. The small number of moving parts in the proposed linear compressor design, along with its insensitivity to dead volume, make it an ideal technology for electronics cooling applications. The results from the sensitivity analysis are used to inform the scalable design procedure for a linear compressor. An improved linear compressor design for electronics cooling is presented with an overall package size of 50.3 mm diameter by 102 mm length and a predicted refrigeration capacity of 200 W.

NOMENCLATURE

A	Area, [m^2]
c	Damping factor, [N sec m^{-1}]
COP	Coefficient of Performance, [-]
D_p	Piston diameter, [m]
f_{res}	Resonant frequency, [Hz]
f	Dry friction coefficient, [-]
g	Leakage gap between piston and cylinder, [m]
h	Enthalpy, [kJ kg^{-1}]
J	Moment of inertia, [kg-m]
k	Stiffness, [N m^{-1}]
L	Length, [m]
$\dot{m}$	massflow, [kg s^{-1}]
M	Mass, [kg]
N	Normal force from piston to cylinder, [N]
n	Polytropic exponent
P	Pressure, [kPa]
$\dot{Q}$	Heat transfer, [W]
$\dot{Q}_{cool}$	Cooling capacity, [W]
r	Radius, [m]
T	Torque, [N-m]
V	Volume, [m^3]
V	Gas velocity, [m s^{-1}]
V_d	Displaced volume, [m^3]
V_{dead}	Dead (clearance) volume, [m^3]
V_r	Volume ratio, [-]
$\dot{W}$	Work over a cycle, [W]
x_d	Distance between piston and valve plate at TDC, [m]
x_p	Instantaneous compressor piston position, [m]
x_s	Compressor stroke, [m]

Greek Letters

β	Angle, [rad]
ϵ	Eccentricity of spring force, [m]
ε_{carnot}	Second law effectiveness, [-]
μ	Oil viscosity, [Pa-s]
η	Efficiency, [-]
ω	Frequency, [rad sec^{-1}]

Subscripts

BDC	Bottom Dead Center
cond	Condenser
cv1	Control volume 1
cv2	Control volume 2
drive	Driving force
eff	Effective
evap	Evaporator
gas	Gas
leak	Leakage
mov	Moving
mech	Mechanical
o,is	Overall isentropic
TDC	Top Dead Center
vol	Volume

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