Clearance Loss Analysis in Linear Compressor with CFD Method

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ABSTRACT

The non-contact seal technology, referred to as clearance seals, is used in linear compressors and promises to maintain the linear compressor for long life operation. The present effort analyzes the aerodynamics in the small clearance package with the commercial Computational Fluid Dynamics (CFD) code Fluent™. Two types of the clearances, the linear seal and the labyrinth seal, are modeled, and the results, such as the pressure and velocity distribution, are described. The clearance loss was quantitatively compared by changing inflation pressure and the gap of the clearance. The results will be used to optimize the design of the linear compressor in the future.

INTRODUCTION

Oil-free piston-cylinder assemblies are widely used in long-life cryocoolers. The cryocooler system is very sensitive to lubricating oil, which is used in traditional rotary compressor. Lubricating oil can deteriorate the cooling condition of the refrigerators; or even destroy the entire cooling system. Due to this disadvantage, engineers and researchers have adopted the oil-free linear compressor. The linear compressor uses clearance seals to pressurize the gap between the cylinder and back volume. Some gas leakage will exist through the clearance seal, especially when there is a shift in the equilibrium location of piston, and this leakage could lead to an unpredictable loss for the compressor

Several authors in the cryocooler field propose mechanisms to minimize the seal loss. For example, when designing the linear compressor, Marquardt and Radebaugh[1] derived a formula to describe the relationship between the seal depth and the PV power loss through the clearance gap. Their equations assumed a uniform gap seal and did not take into account the effects of the labyrinth clearance. If the piston was eccentric to the cylinder, the clearance loss will increase since the flow loss is proportional to the cubic of the clearance seals. Reed and Davey[2] investigated the leakage flow through the clearance gap. In their study, they treated the seal loss as a laminar flow through idealized seal. They assumed the seal type as uniform. Besides,, Spoor and Corey[3] designed the “Anti-drift” piston to overcome the strain pressure problem on the piston, and this method could largely minimize the clearance seal loss. Reed and Bailey[4] established another theoretical equation for the clearance seals. They used the CFD method to substantiate their equation. In the following year, Bailey and Dadd[5] offered the instantaneous mass flow equations through the clearance seal. From the previous research on the clearance seal loss, they established the theoretical equations on the uniform clearance seals. In this research, the description of the detail leakage flow through the two different seals, including linear seal and the labyrinth seal, are computed by CFD .

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This paper presents the new design linear compressor at Zhejiang University, with a moving coil linear motor, linear flexure bearings and clearance seal suspension system. Fig. 1 shows the three dimensional configuration and physical configuration of our linear compressor.

MATHEMATICAL MODEL

The CFD method is used to investigate the detailed mechanism of clearance seal loss. Two types of clearances have been investigated to minimize the leakage flow. In various seal geometries, the straight seal has usually been adopted in previous research studies due to its simplicity. In the sketch, there are two significant points measured in the geometry, the first point is set at the outlet of the compression chamber, which is defined as "Outflow" point; and the other point lies in the back side of the seal, which is defined as the "seal-back" point. These two points are shown in the Figure 2.

Moreover, the comparison of the leakage loss caused by the two kinds of seals, linear clearance seal and labyrinth clearance seal, is discussed in this paper. By fixing other parameters of the mathematical model, only one variable is changed at a time, and the PV work and clearance loss under different working conditions is compared. Specific variations are shown in Table 1.

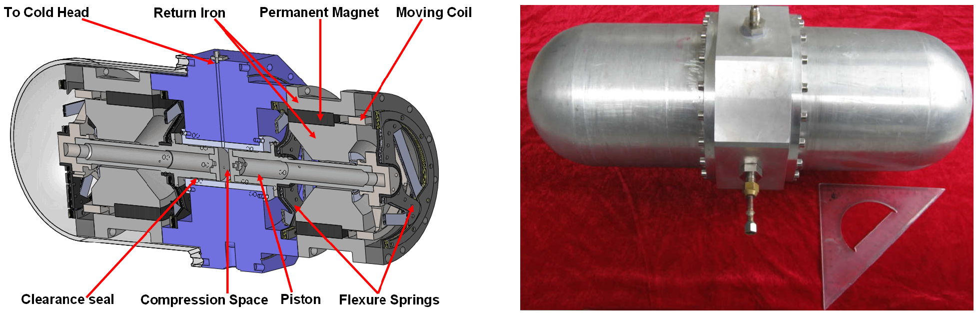
Considering the 2-D flow, neglecting variations in the z-direction, the flow is assumed to be an adiabatic leakage flow. In order to simplify the model, several assumptions are made which are valid with the constraints of small oscillations in the real working conditions. The assumptions are as follows:[3]

1. The wall of the cylinder and piston are adiabatic
2. Piston works in an ideal sinusoidal movement
3. Entire compression process is adiabatic
4. Clearance gap is much smaller than the thermal penetration depth of the working fluid.

The flow in clearance is controlled by the following governing equations.[6]

Navier-Stokes equation: (ignoring variations in the z-direction)

$$\left\{\begin{aligned}&\frac{\partial(\rho u)}{\partial t}+\rho\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}\right)=-\frac{\partial p}{\partial x}+\mu\left[\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{1}{3}\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 v}{\partial x\partial y}\right)\right]\\&\frac{\partial(\rho v)}{\partial t}+\rho\left(u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}\right)=-\frac{\partial p}{\partial y}+\mu\left[\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{1}{3}\left(\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 u}{\partial x\partial y}\right)\right]\end{aligned}\right\} \tag{1}$$



(a) 3-D configuration (b) Physical configuration

Figure 1. Structure of the linear compressor

Table 1. Basic mathematic model parameters of the two types of seals

Seals type	Labyrinth	Linear	——	——
Thickness(mm)	0.02	0.03	0.04	0.05
Charge Pressure(MPa)	2.0	2.5	3.0	3.5
Displacement(mm)	6	6	6	6

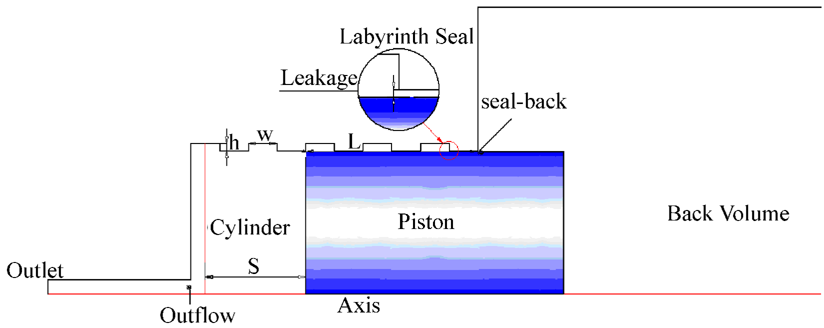


Figure 2. Schematic diagram of labyrinth seal compressor

Energy equation:

$$\frac{\partial(\rho E)}{\partial t}+\nabla\cdot(\vec{v}\rho E)=K\nabla^2 T+\nabla\cdot(\tau_{eff}\cdot\vec{v}) \tag{2}$$

Where E is define as:

$$E=h+\frac{1}{2}\rho v^2-\frac{p}{\rho} \tag{3}$$

Continuity equation:

$$\nabla\cdot(\rho\bar{u})\equiv 0 \tag{4}$$

The equation of state for the ideal gas:

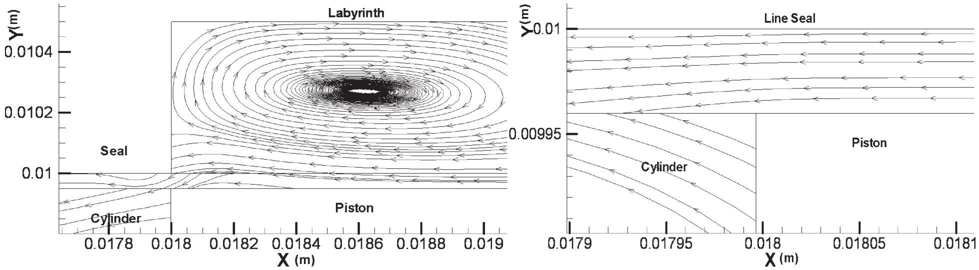
$$pV_{co}=\dot{m}RT \tag{5}$$

In this study, the commercial Computational Fluid Dynamics software, FLUENT™, is used to compute the detail fluid flow pattern. In the models, the reciprocating movement of pistons is realized by the moving mesh technology.

SIMULATION RESULTS

An instructive impression of the leakage fluid flow is gained through the streamline patterns plotted in Figure 3. Obviously, there are two distinct flow patterns between two types of clearances. As shown by inspection of Figure 3(a), it illustrates that the deflection of the main fluid in the labyrinth seal is closely related to the resistance of the leakage. There is an obvious eddy flow in the cavity, which contains the higher pressure than the pressure in the seal. It also reveals that a small separation bubble near the leading edge of the step is formed in the cavity. The corresponding deflection of the bulk flow gives resistance to the leakage. Moreover, the consecutive process of the fluid flow is repeated axially before being discharged out. By careful inspection of Figure 3(b), it discloses that the streamline pattern in the linear seal is straight from the back volume to cylinder.

Aiming to find the mechanism of the seal loss through the clearance, the compression process is treated as an adiabatic process. The total PV power produced by the moving piston is made up of



(a) Labyrinth seal (b) Linear seal

Figure 3. Streamline pattern of the fluid flow

three parts: one is the power transferred to the cylinder defined as PV power, the other is the seal loss due to the leakage flow through the clearance seal, and another loss is the gas spring loss which is ignored in this simulation:

$$\left\langle \dot{W}_{piston} \right\rangle = \left\langle \dot{W}_{pv} \right\rangle + \left\langle \dot{W}_{s} \right\rangle \tag{6}$$

$<W_{pv}>$ is the time average PV power input into the cryocooler. The term $<W_s>$ represents the seal loss caused by the clearance.

$$\left\langle \dot{W}_{PV} \right\rangle = \frac{\pi}{2} f P_l V_{co} \cos\phi \tag{7}$$

The time average PV power is related to the working frequency, the pressure amplitude, the swept volume, and the phase angle between the pressure and volume flow. The loss is conventionally expressed by an analytical solution for leakage flow though an ideal seal gap, which is controlled by the following expression.[2]

$$\left\langle \dot{W}_{seal} \right\rangle = \frac{\pi D t^3 (\Delta p)^2}{96 \mu L_{seal}} \tag{8}$$

From this expression, the clearance loss is mainly related to the thickness and pressure amplitude generated at this tiny gap.

There exists a phase angle ϕ between the volume flow and the pressure wave at the outlet of linear compressor, which is described as the pressure leading the piston position. In the Stirling type cryocooler, ϕ is about 40º to 45º in the experiments; and about 35º to 40º for a pulse tube cryocooler based on experiment.[1] However, in this calculation, there is no cold end such as a displacer or pulse tube assembled in mathematical models. Therefore, the pressure leads the piston position by only a small phase angle which is about 15º at the flowout point. From Figure 4, it can be seen that there is a small phase angle between the mass flow and the pressure through clearance seal, which contains the main clearance loss.

The loss is a function of the thickness of the seal. Based on the enclosure shown in Figure 5, it is easy to find out that the clearance loss decreases when the thickness of the seal is reduced from 0.05 mm

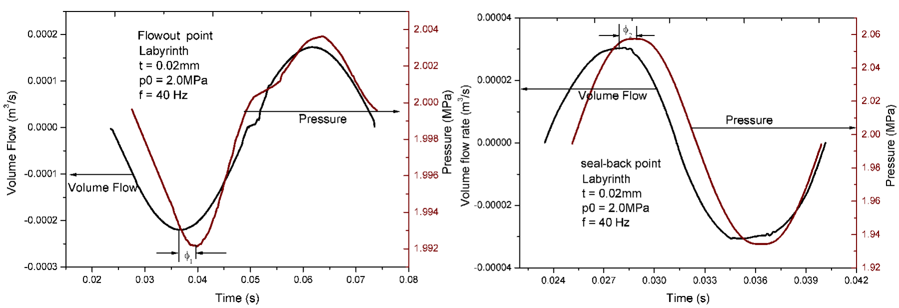


Figure 4. Phase angle between mass flow and pressure wave

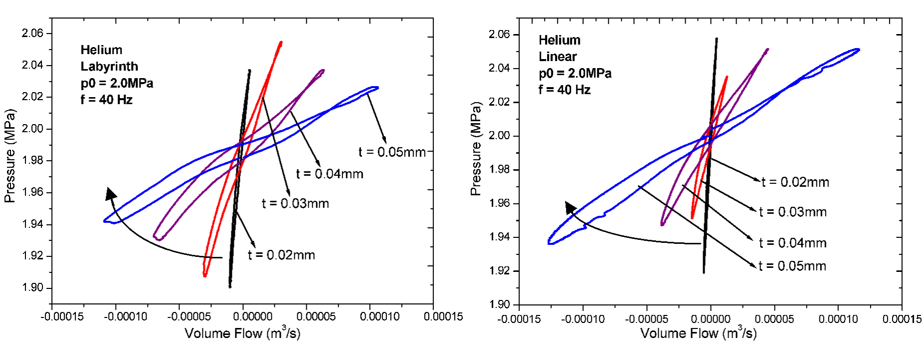


Figure 5. Seal losses related to the thickness of the clearance

to 0.02 mm. The phase angle between the pressure and piston position decreases as the thickness of the clearance rises. At the PV power scheme at the "seal-back" point, the seal loss rotates from vertical direction to the horizon direction in the counterclockwise direction, as the thickness of clearance gets larger.

Figure 6 compares the clearance seal loss and the PV power. Under the same working conditions, the PV work has some dissimilarity owing to different clearances. From Figure 6, when the thickness of the clearance is about 20 microns, the loss from linear clearance is smaller than that of the labyrinth clearance, even though the labyrinth clearance has some cavities. This phenomenon may be caused by the tiny thickness of the clearance gap. What's more, the clearance seal loss generated in the labyrinth seal is a little larger than that generated in linear seal compressor, because the cavities distributed along the labyrinth clearance might be too large compared with the thickness of the clearance. So in the linear compressor, it is better to employ the linear clearance seal to pressurize the gap.

The simulation focuses on how these two clearances influence the PV power. As shown in Figure 7, the diameter of this piston in this case is 20 mm. If the thickness of clearance is less than 0.034 mm, more PV work will be produced by the piston; however, if the thickness of clearance is larger than 0.038 mm, labyrinth seals could prevent a large amount of helium from being transferred into the back chamber, which means labyrinth seals is better than the linear seal in this situation. Linear clearance has less clearance loss if thickness is less than 0.038 mm, which corresponds to the trend in the PV work generated at the outlet. When the thickness of clearance is larger than 0.038 mm, it is better to use labyrinth seal to pressurize the clearance, which may be helpful in other kinds of machine rather than in linear compressor. In order to compare the differences between the theory of clearance loss and the CFD simulating results, plots drawn by these two methods are shown in Figure 7. In Figure 7, the relationship between the linear clearance seal loss and the thickness of clearance is cubic in the CFD model, which is the same as in Eqn. 8. However, the

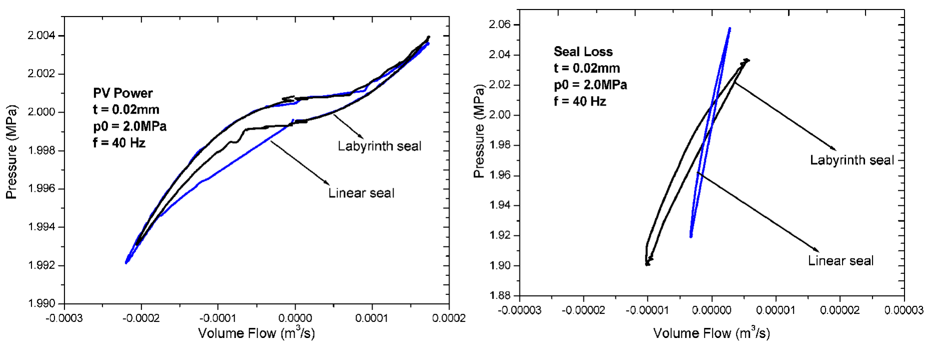


Figure 6. PV power and clearance seal losses produced in three types of clearance seals

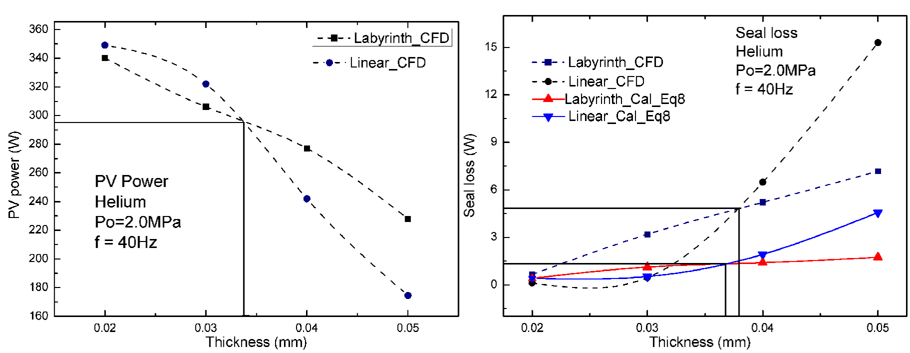


Figure 7. Relationship between thickness and clearance loss

labyrinth clearance seal loss simulated through CFD method is not exactly cubic. The linear seal is better than the labyrinth seal to prevent refrigerants from being injected into the back volume.

Since the relationship between the thickness and the linear clearance seal loss has a cubic exponent, a multitude of PV power will be delivered to the back chamber, as the thickness gets larger. In accordance with Figure 7, labyrinth loss is not exactly the cubic exponent to the clearance thickness. When the thickness is larger than 0.05 mm, there exists a significant loss through this large gap; this loss is larger than the PV work transferred to the cold end. Therefore, the thickness of the clearance has to be less than 0.034 mm in linear clearance seal, with the diameter of the piston is about 20 mm and the compressor swept volume is 20 cm^3 in order to diminish the clearance seal loss.

As the charge pressure ranges from 2.0 MPa to 3.5 MPa, the comparison of the clearance seal loss between Eqn. 8 and the CFD calculation is illustrated in Figure 8. The linear clearance seal loss determined by CFD method is larger than the loss calculated by Eqn. 8; however, the deflection is so small that it can be ignored. Thinking about the labyrinth clearance seal loss, the deviation of the seal loss gets larger when the charge pressure becomes larger, which means some coefficient "k" must be added to Eqn. 8 in order to precisely determine the seal loss caused by the labyrinth seal.

CONCLUSION AND DISCUSSION

Clearance seal loss is often observed in Stirling and pulse tube cryocoolers. Clearance seal is a critical technology in linear compressor, but a loss is introduced by clearance. Nevertheless, there are various types of clearances that could be used in linear compressor. This paper focuses on the common types of clearance seal application. From the mathematic simulation, a few interesting results can be summarized as follows:

1. CFD method is employed to simulate the clearance seal loss in linear compressor. The detail stream line distribution is established along clearances.
2. Two types of clearance are compared to determine the better clearance that could be used in linear compressors. The result shows that if the thickness of the clearance is larger than 0.033 mm, linear seal could provide a smaller clearance loss, as the diameter of the piston is 20 mm. This kind of clearance is also easy to manufacture and assemble. Further research will be placed on discussing the relationship between the length of the clearance seal and seal loss.
3. This simulation analyzes the PV work and clearance seal loss in linear compressor and brings about another calculation method of PV power and clearance seal loss. Facing with the labyrinth clearance seal, the traditional equation might not proper for determining the clearance seal loss in labyrinth seal compressor.

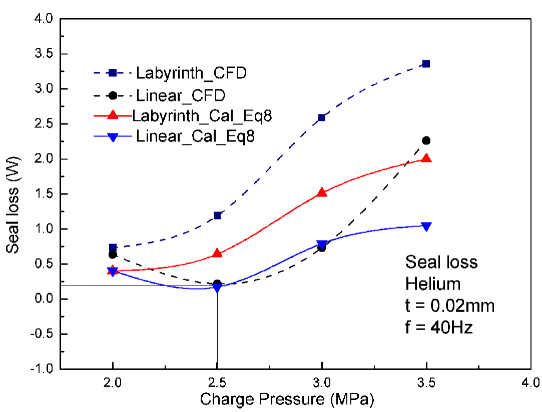


Figure 8. PV power and seal losses related to charge pressure

NOMENCLATURE

D	diameter of the piston
f	driving frequency
L_{seal}	length of the clearance seal
p	instantaneous pressure wave
P_l	amplitude of the dynamic pressure in cylinder
Δp	pressure swing in the seal(peak to peak)
t	thickness of the clearance seal
u	velocity in axial directions
v	velocity in radial directions
V_{co}	compressor swept volume
$<W_{pv}>$	time average PV power
$<W_s>$	seal loss
ρ	density of working fluid
Φ	phase angle between the volume flowand dynamic pressure
μ	fluid dynamic viscosity.
E	total energy of fluid
h	enthalpy for ideal gas
J	diffusion flux of species
τ_{eff}	viscous dissipation coefficient

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