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Performance Comparison between Rolling Piston and Coupled Vane Compressor

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ABSTRACT

In this paper, the performance of the rolling piston compressor and the coupled vane compressor are compared, and differences or similarities are presented and discussed. The comparison is based on one of the available sizes of the rolling piston compressor as used in a room air-conditioner. The extent of the theoretical comparison includes their respective design features, relative sizes, individual flow and frictional energy losses, details of PV diagrams, valve flow analysis and other performance parameters such as "compressor" COP. The coupled vane compressor is a novel compressor recently design with the objective to save raw materials in the production process while matching the performance of current compressors out in the market. It does this through its unique design allows for rotor size to not be limited by the stator size, thus increasing the working chamber to cylinder volume ratio. Through arbitrarily chosen values for their geometrical parameters, the comparison of the two compressors shows that the coupled vane compressor has potential to replace current compressors, thus saving a significant amount of raw materials and cost for compressor manufacturing and creating a more sustainable environment in the future.

NOMENCLATURE

a	R_r/R_c	[-]
A	area	[m^2]
b	distance between the rotor centre and the cylinder centre	[m]
C	specific heat capacity	[J kg^{-1} K^{-1}]
c	damping coefficient	[-]
D	diameter	[m]
E	Young's modulus	[N m^{-2}]
e	eccentric distance	[m]
F	force	[N]
f	motor operating frequency	[Hz]
g	acceleration due to gravity	[m s^{-2}]
h	specific enthalpy	[J kg^{-1}]
h_b	vane height	[m]
I	moment of inertia	[m^4]
k	thermal conductivity	[W m^{-1} K^{-1}]
l	length	[m]
m	mass	[kg]
N	number of items	[-]
O	origin	[-]
P	power	[W]
p	pressure	[Pa]
Q	heat	[J]
q	specific heat	[J kg^{-1}]
R	radius	[m]
Re	Reynolds number	[-]
r	radial coordinate	[m]
s	entropy	[J kg^{-1} K^{-1}]
T	temperature	[K]
t (t_b)	thickness (vane thickness)	[m (m)]
u	specific internal energy	[J kg^{-1}]
V	volume	[m^3]
v	velocity	[m s^{-1}]
v_s	specific volume	[m^3 kg^{-1}]
W	work	[J]
w	width	[m]

Greek symbols

α	contact angle at the leading vane	[rad]
ß	contact angle at the trailing vane	[rad]
ε	eccentricity ratio	[-]
η	efficiency	[-]
λ	friction factor	[-]
μ	dynamic viscosity	[Pa·s]
ρ	density	[kg m^{-3}]
θ	rotation angle	[rad]
ω	angular speed; natural frequency	[rad s^{-1}; Hz]
ζ	damping ratio	[-]

Subscripts

i	inlet (suction) conditions
o	outlet (discharge) conditions
1	upstream conditions
2	downstream conditions
c	cylinder chamber conditions

1. INTRODUCTION

Positive displacement compressors in general are mostly applied in small and medium cooling capacity applications. Rolling piston compressor is the most widely used positive displacement rotary compressor in the world today. It is particularly popular in household air-conditioners and those small and medium size cooling load applications, generally from fractional kW up to 15 kW of cooling capacity. Its annual production rate has been increasing, and for the past few years, it has been hovering around 170 to nearly 200 million pieces annually, making it the most produced compressor in the market. As compared to other types of positive displacement compressors, it has the inherent characteristics of a simple geometry, having few parts, low vibration and being compact.

Due to the large annual production volume of the rotary compressors, large amounts of material go towards making the compressor and most of these materials are metallic materials. If the general weight of a compressor is taken as 20 kg, 3-4 billion kg of metal will be used annually just for fabricating the rotary compressors alone. In the interest of saving materials, one of the newly introduced rotary compressor design, namely coupled-vane compressor (CVC) was born. (Ooi and Shakya, 2018a) For the same cooling capacity, this new compressor needs about half, if not less, of the material in fabricating it, and yet has about the same number of parts, if not fewer.

As a result of the innovative design feature, this CVC allows a rotor to cylinder ratio of less than 0.4, instead of greater 0.7 for all the existing rotary compressors, making it the most compact rotary compressor design available.

This report aims to study and compare the differences in the design and working principles of the rolling piston compressor with that of the CVC. The mathematical model used in the simulation programme to predict the performance of the compressors will also be briefly presented and discussed. The results of the simulation programme following certain operating parameters are compare and presented, including the frictional energy losses, PV diagrams, valve flow analysis, and other performance parameters.

2. ROLLING PISTON COMPRESSOR

Rolling piston compressors are known for having high efficiency and producing low noise. Because of this, they are widely found in domestic refrigeration and air-conditioning units (Matsuzaka and Nagatomo, 1982).

2.1 Design

A rolling piston consists of a rotor which is being housed in a cylinder, with its centre being eccentric relative to the cylinder centre. The compressor operates through the rotation of the rotor along the cylinder walls which moves the vane, causing it to reciprocate within its slot found along the cylinder walls. This vane separates the working chamber into its suction and compressor chambers.

These design factors of the rolling piston result it in having low vibrational problems as its parts are well-balanced throughout each compression cycle. However, there are multiple rubbing components and leakage paths which might lead to losses in the overall compressor efficiency (Teh and Ooi, 2009).

2.2 Mathematical Model

A mathematical model can be formulated by including heat transfer, valve dynamics, and the thermodynamics and flow aspects. To go more in-depth, stress and mechanical losses can also be considered. Similar to a previous study done (Ooi, 2004), the mathematical model referred to in this project does not include heat transfer for ease of computation. Results produced in that simulation through the omission of heat transfer has been shown to be comparable to that of actual measured results and is thus concluded to be suitable for use in this evaluation.

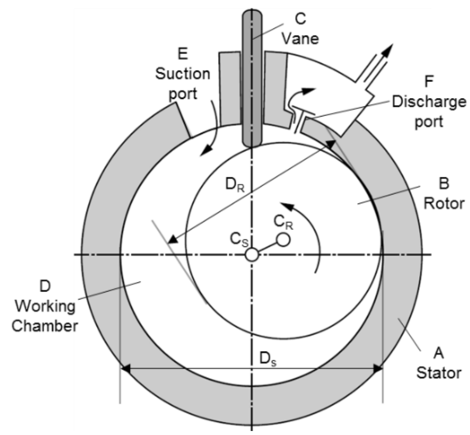

Figure 1: Rolling Piston Compressor Schematics (Soedel, 2007)

2.2.1 Geometrical Model: Figure 1 shows the schematics of the rolling piston compressor. Its working volume is split into the compression and suction chambers. The total working chamber volume is the volume found between the stator, rotor, and piston, bound by the two end plates of the compressor. This can be calculated as follows:

$$
\begin{aligned}
V(\theta) = \frac{IR_c{}^2}{2}\Bigg[&(1-a^2)\theta - \frac{(1-a^2)}{2}\sin 2\theta \\
&- a^2 \sin^{-1}\left(\left\{\frac{1}{a}-1\right\}\sin\theta\right) - a(1-a)\sin\theta\,\sqrt{1-\left(\frac{1}{a}-1\right)^2\sin^2\theta}\Bigg] \\
&- \frac{t_b h_b}{2} R_c\left[1-(1-a)\cos\theta - \sqrt{(1-a)^2\cos^2\theta + 2a - 1}\right]
\end{aligned} \tag{1}
$$

2.2.2 Thermodynamic Processes: The First Law of thermodynamics can be used to determine the variation of pressure, mass, and temperature of the working fluid through the suction, expansion, compression, and discharge processes. Gas relations properties and equations for the conservation of mass are also used. The above variables can be connected through these equations.

$$
\frac{dT_c}{dt} = \frac{\frac{dv_s}{dt}\left[\left(\frac{\partial h_c}{\partial v_s}\right)_T - v_c\left(\frac{\partial p_c}{\partial v_s}\right)_T\right] - \frac{1}{m_c}\left[\frac{dQ}{dt} + \sum \frac{dm_i}{dt}(h_i - h_c) - \sum \frac{dm_o}{dt}(h_o - h_c)\right]}{\left[v_c\left(\frac{\partial p_c}{\partial T_c}\right)_{v_s} - \left(\frac{\partial h_c}{\partial T_c}\right)_{v_s}\right]} \tag{2}
$$

$$
\frac{dp_c}{dt} = \frac{dT_c}{dt}\left(\frac{\partial p_c}{\partial T_c}\right)_{v_s} + \frac{dv_s}{dt}\left(\frac{\partial p_c}{\partial v_s}\right)_T \tag{3}
$$

$$
\sum \frac{dm_i}{dt} - \sum \frac{dm_o}{dt} = \frac{dm_c}{dt} \tag{4}
$$

In-cylinder convection analysis can be conducted using convective heat transfer model to determine the value of Q.

3. COUPLED VANE COMPRESSOR

The coupled vane compressor is a novel compressor designed to improve current compressor designs through the reduction in the material used for its manufacturing and increasing the working volume for a certain size of compressor but reducing the need for the rotors to be of a minimum size as compared to the cylinders (Ooi and Shakya, 2018b).

3.1 Design

The coupled vane compressor is a novel compressor designed with a twin-vane system which run diametrically through the rotors, and slides in and out of the rotor and along each other during the operation of the compressor. Pressure from the discharge chamber will balance with the centrifugal force of the vane’s rotation and the pressure from the suction chamber to keep the vane tips in contact with the wall.

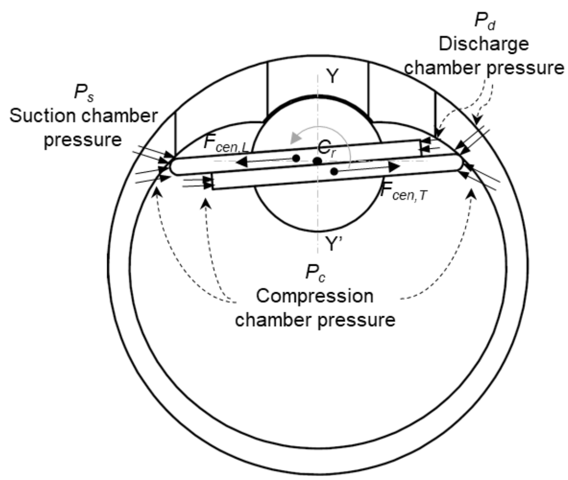

Figure 2: Coupled Vane Compressor Schematics (Ooi and Shakya, 2018c)

3.2 Mathematical Model

Similar to the simulation of the rolling piston compressor, the simulation for the CVC will include a mathematical model through derivations of the working chamber geometry, working fluid thermodynamics, suction and discharge port primary flows, and secondary leakage flow occurring at the internal clearances. Heat transfer is also excluded in this case for simple computations. Details the mathematical model and its derivation have been done by Ooi and Shakya (2018).

3.2.1 Geometrical Model: Figure 2 shows the schematics of the CVC. Similar to the rolling piston compressor, its working volume can be split into the compression and suction chambers. The total working volume, however, defers from the rolling piston due to their different design geometries. For the CVC, the working volume can be calculated as follows:

$$V_c(\theta_r) = \frac{l_c}{2}\int_0^{\theta_r}\left(r(\theta_r)\right)^2 d\theta_r - l_c\left(\pi R_r^2 \times \frac{\theta_r}{2\pi}\right)$$
$$= \frac{l_c}{2}\left[R_c^2\theta_r + \frac{b^2}{2}sin(2\theta_r) - bsin\theta_r\sqrt{R_c^2-(b\sin\theta_r)^2} - R_c^2\tan^{-1}\left(\frac{bsin\theta_r}{\sqrt{R_c^2-(b\sin\theta_r)^2}}\right)\right] - l_c\left(\pi R_r^2 \times \frac{\theta_r}{2\pi}\right) \tag{5}$$

3.2.2 Thermodynamic Processes: The complete working cycle of the CVC includes the suction, compression, and discharge processes. A control volume is assumed for the prediction of the thermodynamic properties of the working fluid. The following equations are used to determine the thermodynamic properties:

$$\sum \dot{m}_{in}(u+pv)_{in} - \sum \dot{m}_{out}(u+pv)_{out} + \dot{Q} - \dot{W} = [\dot{m}(u) + m(\dot{u})]_{system} \tag{6}$$

$$\frac{dp}{dt} = \left(\frac{\partial p}{\partial T}\right)_{\rho}\frac{dT}{dt} + \left(\frac{\partial p}{\partial \rho}\right)_{T}\frac{d\rho}{dt} \tag{7}$$

$$\dot{m}_{cv} = \dot{m}_{in} + \dot{m}_{leak,in} - \dot{m}_{out} - \dot{m}_{leak,out} \tag{8}$$

4. SIMULATION RESULT

The simulation is carried out with the AHRI Standard 540, 2020 Standard for Performance Rating of Positive Displacement Refrigerant Compressors. Details of which are briefly covered as follows:

Table 1: Compressor Testing Standards

Compression Cycle			Low Side		High Side – Subcritical	
Application	Rating Test Point	Cycle Type	Suction Dew Point Temperature, °C	Superheat, K (or Return Gas Temperature), °C	Discharge Dew Point Temperature, °C	Condenser Exit Sub Cooling , K
Refrigeration	Medium	Subcritical	-10	10 (20)	45	0

The above ratings are tested at 35°C ambient temperature surrounding the compressor. Further details of the testing standards can be found in the AHRI Standard 540 – 2020.

4.1 Volume Tested

The volume tested for each compressor is equal to bring them towards a fairer comparison. This is done by ensuring that the maximum working chamber volume per cycle is the same for each compressor. This maximum volume occurs at theta equal to 360° for the rolling piston compressor and 270° for the CVC.

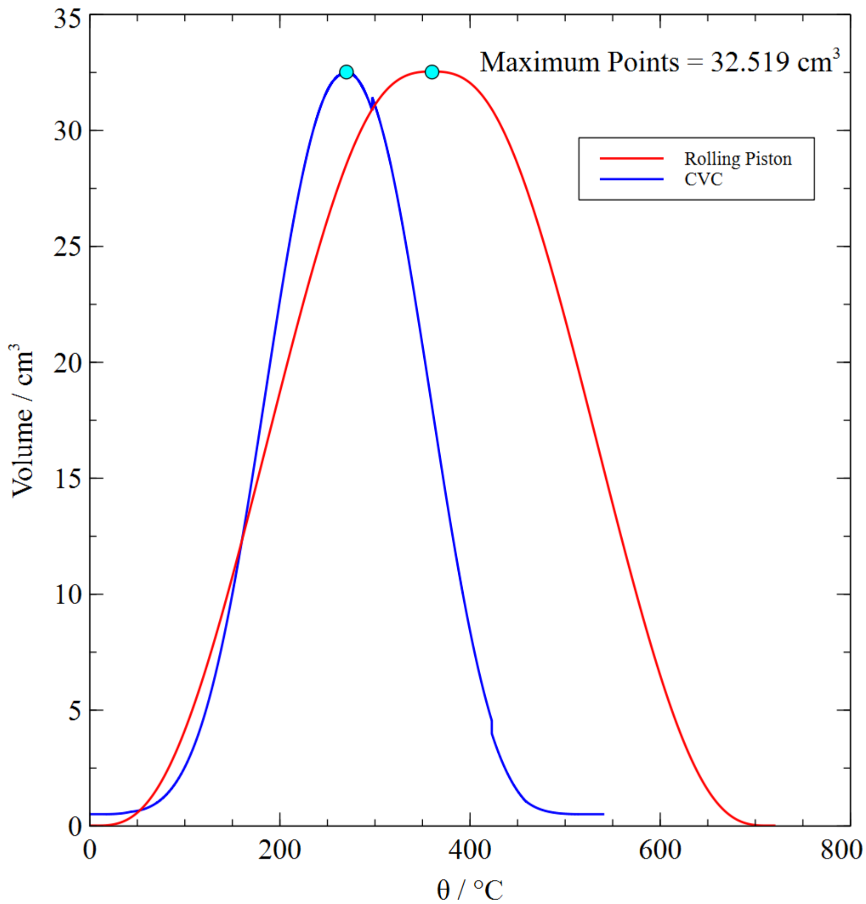

Figure 2: Volume of Working Chamber for Rolling Piston Compressor and CVC

The above can be achieved by ensuring that the geometric parameters for both compressors results in the same volume.

Table 2: Geometric Parameters of Rolling Piston Compressor

Geometric Parameter (RP)	Value
Cylinder Radius / mm	58.020
Rotor Radius / mm	46.780
Working Chamber Height / mm	35.150
Bearing L/S / mm	22.250
Suction Port Diameter / mm	16.0
Discharge Port Diameter / mm	7.0
Working Chamber Maximum Vol. / cm^3	32.519

Table 3: Geometric Parameters of Coupled Vane Compressor

Geometric Parameter (CVC)	Value
Cylinder Radius / mm	32.500
Rotor Radius / mm	20.250
Working Chamber Height / mm	18.759
Suction Port Diameter / mm	16.0
Discharge Port Diameter / mm	7.0
Working Chamber Maximum Vol. / cm^3	32.519

The cross-sectional dimensions being the main variable factors which affect their performance. Suction and discharge valve sizes in both the rolling piston and CVC have the same value.

The rolling piston compressor is running at 3000 rpm while the CVC is running at 4000 rpm. This is due to each compression cycle of the CVC going through 1.333 times the angle of a full compressor cycle of the rolling piston compressor. This variation in the rotation rate is to ensure that both compressors compress the same amount of air per unit time.

4.2 Fluid Tested

R134a is selected for testing with both compressors through the simulation programme.

5. RESULTS AND DISCUSSIONS

Results from the simulation studies are presented below. It is worth noting that the simulation programmes, although based on the mathematical models of the working principles of the compressors, is not able to completely encompass all the details of a compressor's operation. They have been compared with results of actual physical testing and have shown a reasonable degree of accuracy for them to be used as an estimate of the performance of the compressor. Thus, results obtained should be used as a gauge and not of the actual performance of the compressor.

5.1 Results

The following shows the results for testing of the compressor with R134a as the working fluid:

Table 4: Mechanical Analysis of Rolling Piston Compressor using R134a

Frictional power losses	W	Contribution (%)
Loss due to vane side reactions	29.952	29.955
Loss due to vane tip roller force	6.432	6.433
Loss due to roller and eccentric friction	31.863	31.866
Loss due to roller and cylinder head friction	0.019	0.019
Loss due to eccentric and cylinder head faces	0.750	0.750
Loss due to bearing S and shaft	11.308	11.309
Loss due to bearing L and shaft	19.666	19.668
Total friction power	99.990	100%

As can be seen from the rolling piston, a large portion (around 60%) of the losses occur as a result of roller and eccentric friction.

Table 5: Mechanical Analysis of CVC using R134a

Frictional energy losses (180deg)	J	Contribution (%)
Loss due to vane tip 1	1.174	32.062
Loss due to vane tip 2	1.620	44.242
Loss between vane and rotor	0.182	4.957
Loss between vane faces	0.097	2.641
Loss due to rotor endface fluid	0.191	5.218
Loss due to upper bearing	0.182	4.960
Loss due to lower bearing	0.217	5.919
Total friction power	3.663	100%

Simulation studies done on the CVC running at 3000rpm and using stainless steel 17-7 PH vanes resulted in the main contribution for frictional loss being between the vane tip and the cylinder, that is almost 80% of total frictional loss.

The P-V diagrams for both simulations carried out are as shown:

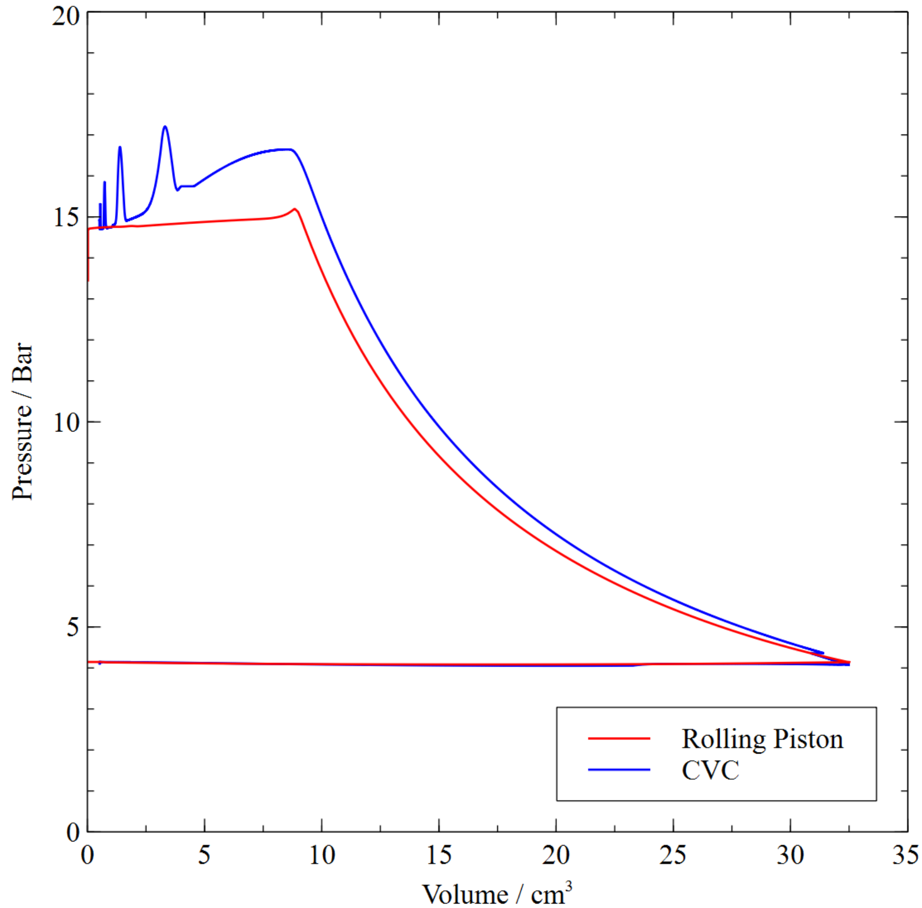

Figure 3: P-V Diagrams for Rolling Piston Compressor and CVC

As can be seen from the diagrams, both compressors undergo similar variations in their pressure and volume throughout their compressor cycles. Because of the difference in rotation rates, the CVC goes through more of such compression cycles per unit time.

Additional performance summary of both the compressors are presented in the table below:

Table 6: Performance Summary of each Compressor

Compressor Type	COP	Mechanical Efficiency	Volumetric Efficiency
Rolling Piston	3.297	89.78%	97.92%
Coupled Vane	2.351	67.29%	95.68%

5.2 Discussion

There are further improvements that can be done to ensure a better comparison between these two compressors. In the simulation studies done, cylinder radius, rotor radius, and vane height were the main geometric parameters varied. In future studies done, more parameter can be altered such as the vane thickness, length, and clearance values.

In addition, rotation rates of each compressor can be equal to ensure a fairer comparison. This can be done by changing the working chamber volume of the compressors with resulting overall volume compressed per unit time remaining the same. The simulations can also be repeated with other refrigerant such as R32 or R1234yf or other fluids to test the consistency of the results.

Actual experiments can also be conducted to ensure that the results from the simulation can be replicated and that actual results do not suffer too large a difference when compared to the results of each compressor's mathematical model.

6. CONCLUSIONS

It is difficult to compare two different compressors due to their unique design and the differing required operating parameters for each of them. The overall results show that the dimensions and thus raw material required for the CVC is smaller than that of the rolling piston compressor for the same amount of volume being compressed per unit time. This is mainly due to the relatively smaller size of the rotor compared to the cylinder for the CVC, in this case 32.50mm for cylinder size. The rolling piston compressor has a rotor to cylinder radius ratio of 58.02mm.

The CVC suffers in some of its overall performance parameters, with a lower COP, mechanical efficiency, and volumetric efficiency as compared to the rolling piston compressor, which means that more power is required by the CVC to carry out the same compression process. The most significant difference lies in the mechanical efficiency, having a value of 67.29% for the CVC, which pales in comparison to the 89.78% for the rolling piston compressor as shown in Table 6. This is likely a result of the significant frictional losses between the vane tips and inner cylinder walls. To counter this, the vane dimensions, or angular velocity of the compressor can be varied to ensure that the force balance at the vane tip is more equal. If successful, less reaction force will be required by the cylinder inner walls to counter the centrifugal forces of the vanes, resulting in a reduction of the frictional forces between them as well. In addition to this, a more suitable lubrication method can be chosen to further reduce friction. With less frictional losses from the vane tips, improvements to the mechanical efficiency of the CVC will be observed.

It is worth noting that the CVC is a novel compressor and is therefore still in its early phase of development. Many improvements can still be made to its design and thus overall efficiency - a department where it is found lacking when compared to the rolling piston. For the CVC, the absence of too large a restriction on its rotor size allows for a larger working chamber volume per cylinder size. This proves a significant advantage in its design and possible future potential.

It is possible to conduct optimisation studies on each compressor and comparing their performance thereafter to ensure a fairer comparison of the two as opposed to arbitrarily chosen values for their geometrical parameters.

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