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# Inception of the Revolving Vane Compressor

**By**

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**Abstract:**

This paper presents a step-by-step design details leading to the inception of a new rotary compressor called Revolving Vane compressor (the RV). This compressor was conceived with the objective to counter all the basic design limitations present in the existing rolling piston compressor (the RP). The outcome is the RV compressor. As compared to the rolling piston design, this new compressor has the following advantages: fewer components; lower frictional losses and thus a higher mechanical efficiency, a higher volumetric efficiency and a simpler construction. The RV has a rotating cylinder and a vane which is rigidly fixed to the driving component, in this case, the cylinder. A mathematical model has been formulated to understand the performance of the compressor and a prototype compressor has also been fabricated and tested to proof the design concept.

**Keywords:** prototype compressor

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## 1. Existing Arts

Compressors have been used for hundreds of years. They are used in air compression, heating, refrigeration and air-conditioning. According to Japan Air conditioning, heating & Refrigeration News [1], there were about 295 million compressors produced for applications in refrigeration and air-conditioning. The main compressor types are reciprocating compressors (60% by volume) and the rotary sliding vane compressors (35% by volume). The remaining are scroll and the screw compressors. The reciprocating compressors are mainly used in refrigerators, the power range of which is normally less than 0.5 kW while the rolling piston compressors are mainly used for room air conditioners, the power range of which is up to 2.5 kW. The scrolls are mainly used in higher power ranges of air-conditioning applications; those in the range of 2.5-7.5kW. These three types of compressor make up almost the entire US$21 billion world compressor market for refrigerators and air-conditioners.

The rolling piston compressor (the RP) is probably the one that is most widely used yet simplest in its geometrical construction, cheapest to produce and easiest to fabricate. Refer to Figure 1, it has only five components: a cylinder, a roller, a shaft integrated with an eccentric, a vane and a spring. This popularity of the RP over the more established reciprocating compressor is mainly due to its inherent characteristics as a rotary machine which are simpler in construction, more compactness, higher operational speeds, lower vibration and lower noise. Today, it has used in most of the room air conditioners, the place of which was previously dominated by the reciprocating compressors.

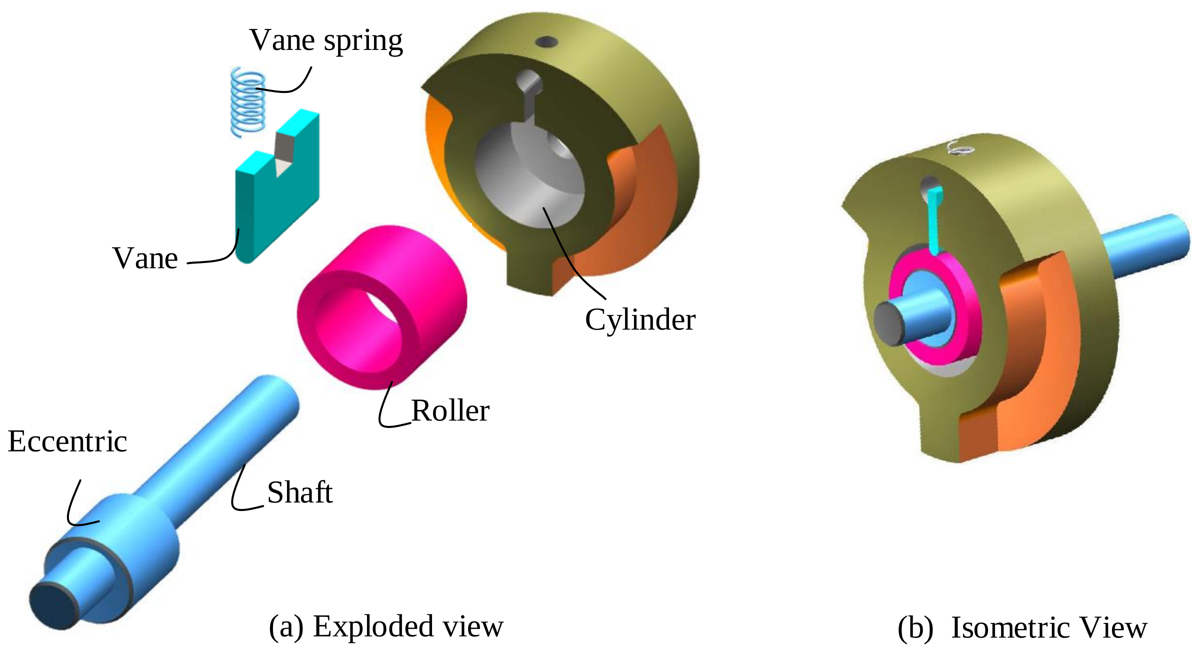



Figure 1 Five major components of the rolling piston compressor

However, due to its high speeds of operation and the characteristics of its basic mechanism design, the RP produces a higher frictional loss as compared to its reciprocating counterpart. In the RP, frictional losses occur at the following six rubbing locations [2]:

(i) roller and eccentric,
(ii) vane and vane slot,
(iii) vane-tip and roller,
(iv) endfaces of eccentric and cylinder,
(v) endfaces of roller and cylinder, and
(vi) shaft bearings.

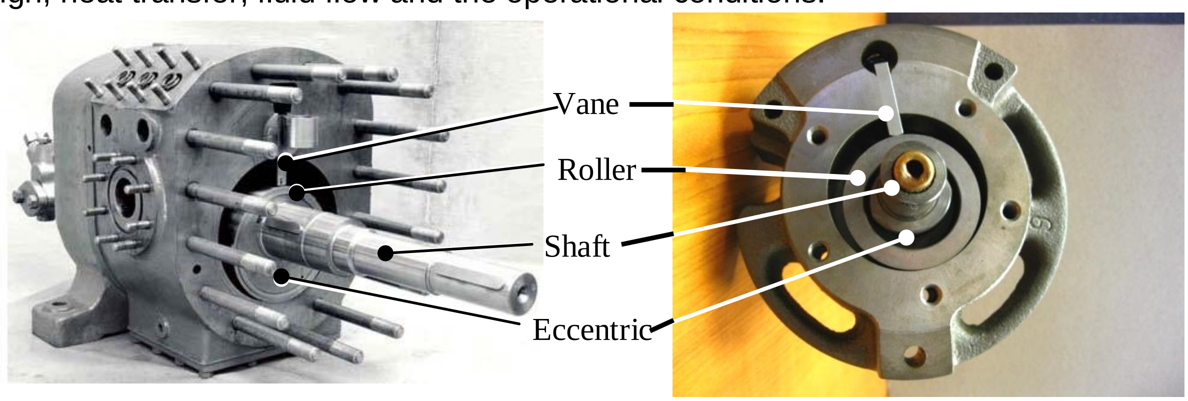
Since the early 1970s, significant research and developmental efforts from industries, research centres and universities [2-11] have significantly improved the efficiency of this compressor. Today, we can expect any further improvement in energy efficiency from this machine to be marginal, if not impossible. The frictional losses occurring in this compressor also limit its compactness and the energy efficiency level.

In an attempt to further improve the energy efficiency of the rolling piston compressor, efforts have been made to improve the energy efficiency by eliminating or reduce the frictional losses of this compressor. The following sections present a step-by-step design approach by which the frictional losses at those six locations are reduced or completely eliminated. The result is the novel Revolving Vane compressor (the RV).

## 2. Inception of the RV

### 2.1 Introduction

W.S.E. Rollaf [12] introduced the rolling piston compressor (the RP) in 1920. Figure 2(a) shows the 1930 version of the RP which was used in a heat pump system [12]; the recent version is shown in Figure 2(b). From these two figures, it can be seen that its basic design remains unchanged. During the operation, the eccentric which forms an integral part of the drive shaft rotates the roller and causes the volume trapped within the inner wall of the cylinder, the rotor and the vane to vary and hence results in a complete compressor cycle. The performance of the machine is determined by the balanced among its mechanical design, heat transfer, fluid flow and the operational conditions.




(a) Rotasco Compressor of the City Hall Heat Pump in 1930 [12]

(b) A Recent Version of the Rolling Piston Compressor

**Figure 2 A** 1930 [12] and a recent version of the rolling piston

## 2.2 From Rolling Piston (the RP) to Revolving Vane design (The RV)

This section presents proposed design approach in which the limitations of the RP are eliminated or reduced one by one in order to arrive at the inception of the new revolving vane compressor. Table 1 shows the frictional analysis of a RP used in refrigerator [3]. The actual contribution of each frictional loss component to the total frictional loss in a compressor design depends on the operational conditions and the detailed design of compressor parts, dimensions and tolerances. For the detailed theoretical an experimental analysis of the RP, readers can refer to refs [2-6].

**Table 1** Mechanical Analysis of the RP compressor
(working fluid: R134a, compressor shaft input power:155 W, motor efficiency: 80%, mechanical efficiency: 62.24% and COP:1.58) [3].

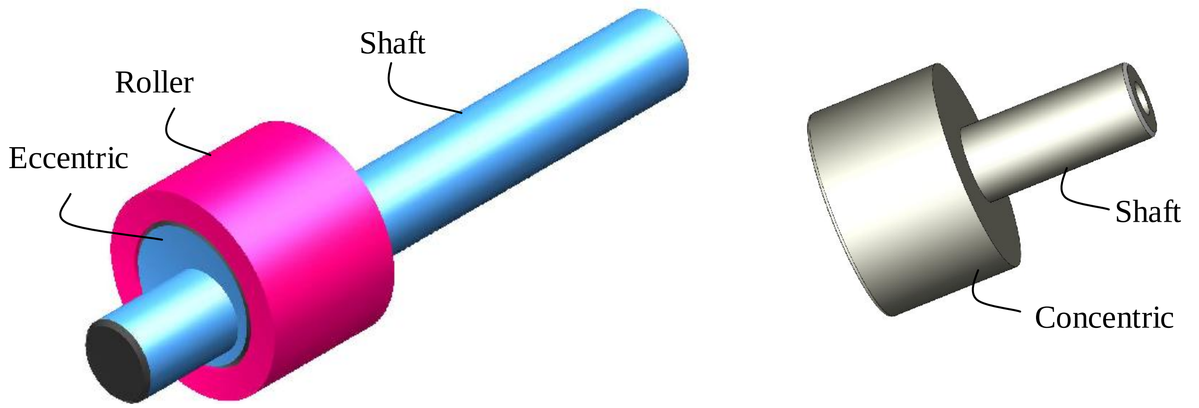
| Frictional losses | Contribution (%) |
|---|---|
| Loss due to vane side reactions | 0.39% |
| Loss due to vane tip roller force | 0.87% |
| Loss due to roller and eccentric friction | 45.67% |
| Loss due to roller and cylinder endfaces | 8.84% |
| Loss due to eccentric and cylinder endfaces | 17.32% |
| Loss due to bearing S and shaft | 6.83% |
| Loss due to bearing L and shaft | 20.07% |
| Total friction power | 100.0% |

(i) From roller and eccentric in the RP to a concentric in the RV

The rubbing between the roller and the eccentric in the RP is due mainly to the contact between these two components, see Figure 3(a). The existence of the roller in the RP was due to the need to reduce the vane tip friction loss. During the operation, the roller is always in contact with the vane tip on its external surface and the eccentric in its internal surface. If the friction at the vane tip is higher, the roller will rotate relative to the eccentric, however if the friction between the inner surface of the roller and the eccentric is higher, then the roller will rub against the vane tip. This way, it allows a lower friction to occur at the roller and at the same time, reduces the vane tip friction. Without the roller the vane tip is constantly rubbing against the eccentric. The latter situation will cause significant wear at the vane tip and will hamper the proper operation of the RP.

Over the past many years, through continuous efforts in design improvement, the frictional loss caused by the rubbing at the interface of these components has greatly reduced. Recesses have been introduced to reduce the contact areas, lubricating oil grooves have also be designed to lead lubricant to the rubbing areas. However, total elimination of the friction is not impossible and this can be achieved by using only one component instead of two. Figure 3(b) shows that for the RV, there is no roller but the shaft and the eccentric present in the RP compressor are now integrated into just one component called “concentric”.

In Table 2, for the case shown [3], this frictional loss is a significant 45% of the total frictional loss, and this will be nil in the case of the RV. Additionally, the RV design also results in a reduction of one component and at the same time eliminate the potential vibration caused by the imbalanced rotating inertia due to the existence of the eccentric in RP.




(a) Roller and eccentric-shaft of RP compressor are two separate components

(b) A Concentric and shaft of the RV compressor is one component

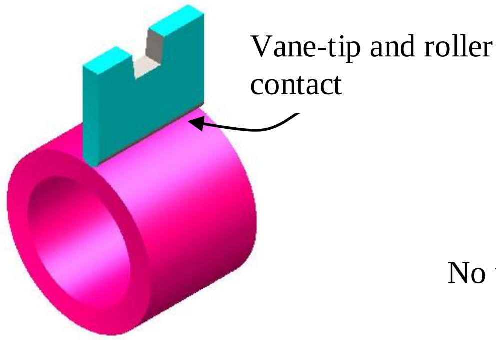
Figure 3 Roller, Eccentric and shaft in the RP become a concentric-shaft in the RV

(ii) Elimination of vane tip – roller friction

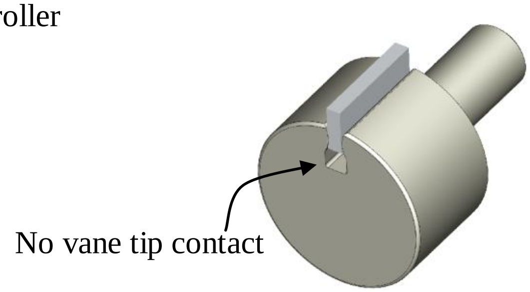
In the RP, the vane is constantly in contact with the roller and hence the frictional loss at the vane tip is unavoidable, see Figure 4(a). The vane must thus be hardened to reduce wear. This vane tip-roller arrangement in RP also constrains the maximum speed of operation of the compressor. Though Table 2 shows that the contribution from vane tip friction is not very significant, however, as mentioned before, the actual amount of the frictional loss is actually dependent on the size of the vane and its thickness, as well as the operational conditions, as the back of the vane is exposed to the discharge pressure. For RP compressor [3, 4], this friction loss $L_v$ at the vane tip is proportional to

$$L_v \propto (t_v, l_v, P_d) \quad (1)$$

where $L_v$ is the vane tip frictional loss, $t_v$, $l_v$ and $P_d$ are vane thickness, vane length and the discharge pressure of the compressor respectively; hence a small vane leads to a low vane tip frictional loss, and vice versa. To eliminate the friction and wear at the vane tip and to enable the compressor to operate at higher speeds, the vane tip must not be in contact with the roller at all time. All these limitations can be overcome by placing the vane into the vane slot which is cut on the concentric in the RV and a provision is made such that the vane tip will not be contact with the base of the vane slot, see Figure 4(b).




(a) Vane-tip roller rubs against each other in RP



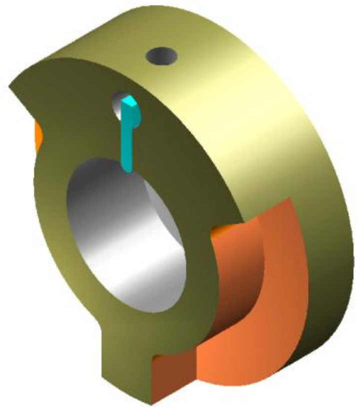

(b) The vane is in the slot in concentric and no vane tip contact in RV

**Figure 4** Vane Rubs Against Roller in RP but No Vane Tip Contact in RV.

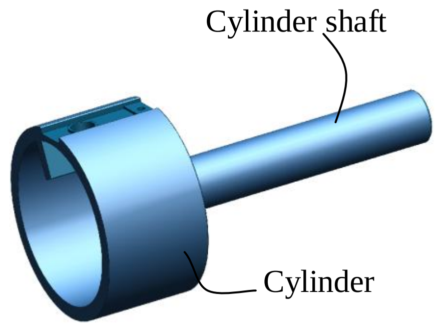
(iii) <u>Lower vane slot friction in the RV</u>

For the vane slot friction, let's consider the case of the RP before examining that of the RV. In the RP, there exists friction between the vane sides and the slot, and the most unwanted attribute is that the magnitude of the vane slot friction is directly proportional to the gas pressure force acting on the vane. The gas pressure force is caused by the pressure differential across the vane. The dependency of the frictional loss at the vane slot to this gas pressure force is due to fact that the vane is allowed to move freely within the slot. The larger the gas pressure force acting on the vane side is, the larger the vane side-slot contact force, and hence the higher the frictional loss at the vane slot. Figure 5(a) shows that the vane in the RP is allowed to move freely in the slot.

Since the vane separates the suction and the compression chambers, there is always a significant pressure difference across the vane which may result in a substantial vane slot friction loss, depending on the absolute value of the gas pressure difference across the vane and the size of the vane. To overcome this problem, in the RV, the vane is rigidly fixed to the cylinder, and if the cylinder is driven by the motor, the frictional force in the vane slot as a result of the gas pressure across the vane is no longer present and thus the vane side-slot contact force is no longer dependent on the gas pressure acting on the vane. Figure 5(b) shows the vane is rigidly fixed to the cylinder in the RV. Doing so causes the vane-slot contact force to be completely independent of the gas pressure force, if the motor is connected to the cylinder shaft, as shown in Figure 5(b). This can be an important advantageous factor for the RV, especially when an excessively large pressure difference exists across the vane, such as those occur in the $CO_2$ refrigeration compressors, where the pressure difference is commonly in excess of 7 MPa [13].



(a) The vane is allowed to move freely: "sideway" and radially in the RP




(b) The vane is rigidly fixed on to the cylinder in the RV

**Figure 5** The vane rubs against the slot in RP and is worsen by the large pressure differential cross the vane. In the RV the pressure differential has no effects on the vane-slot friction.

For the RP [3], the frictional loss between the vane sides and the vane slot, $L_{vs,rp}$ is proportional to pressure differential across the vane, i.e.

$$L_{vs,rp} \propto (\Delta P) \tag{2}$$

where $L_{vs,rp}$ and $\Delta P$ are vane-slot friction loss for RP and the pressure differential across the vane. While for the RV, the vane-slot friction loss, $L_{vs,rv}$, is proportional to rotating inertia of the driven component, which is

$$L_{vs,rv} \propto (I_{ro}) \tag{3}$$

where $I_{ro}$ is the rotating inertia of the concentric-shaft. It can be seen from eqn (3) that the vane side frictional loss of the RV is independent on the pressure differential across the vane [14]. Calculations show that $L_{vs,rv}$ is generally lower than $L_{vs,rp}$ for compressor with similar capacity and working at the same operational conditions.

(vii) <u>Lower cylinder end faces friction in the RV</u>

In the RP, as the rotor rotates the endfaces of the roller and eccentric rub against the stationary cylinder endfaces and hence results in a significant energy loss. This rubbing at the endfaces is caused by the "rotating" roller and eccentric rub against the "stationary" cylinder. To overcome this friction, the cylinder in the RV is now made to rotate together with the rotor. As mentioned previously, the motor is connected to the cylinder shaft to rotate the cylinder and since the vane is rigidly fixed to the cylinder it then rotates the rotor. Since both rotor and cylinder rotate together at the same average speed, the relative velocity between these two components is low and hence the endface friction is significantly reduced. The relationship for the endface friction for the RV [14], $L_{ef}$ is dependent on the relative velocity $V$ between the rotor and the cylinder as shown in eqn (4).

$$L_{ef} \propto (V^2) \tag{4}$$

A lower $V$ will thus result in a lower frictional loss.

(viii) Significantly lower shaft bearing friction in RV than that in RP

The shaft bearing is one of the most crucial design factors in rotary compressor. For the case of RP or RV the shaft bearing load is general transmitted from the pressure force to the shaft bearing. The bearing force $F_{br}$ varies as the compressor operates and is proportional to the size of the cylinder and can be given as:

$$F_{br} \propto (h_c \times D_c) \tag{5}$$

where $h_c$ and the $D_c$ are the height of the cylinder and the diameter of the cylinder respectively. The determination of the size of the cylinder is dependent on the cooling capacity of the compressor. One can choose to have a thin and long cylinder or a fat and short cylinder to have the same cooling capacity. However, this choice is constrained by many factors that affect the performance of the compressor. In general a short and fat configuration of a cylinder will always give a lower shaft bearing force but with a larger vane slot friction. The latter is caused by a larger gas pressure force acting on the vane. However, in the RV as the vane slot frictional losses is independent of the pressure differential, it is thus possible to have a fat and short cylinder in order to have a lower the bearing force without affecting the vane slot friction. All in all, not only the RV has fewer frictional areas, but each of the frictional losses is lower than those in the RP, as explained above. Table 3 shows that the RV has a highest predicted mechanical efficiency.

### 2.3 Suction and discharge ports design considerations for the RV

In the RV, the cylinder is constantly rotating, the discharge port, which is located on the cylinder, also rotates together with the cylinder. The analysis [15] has shown that the centrifugal effects acting on the rotating valve reed softens the reed and eases the valve opening, and thus brings benefits by reducing the discharge valve loss.

For the suction port, since one of its ends needs to be connected to the suction pipe linking the evaporator, this end of the suction port must be non-rotating. But since in the RV, all parts rotate, the external-end of the suction inlet must be located at the compressor housing, similar to all other hermetic type compressors. The inner-end of the suction inlet has to be located at the centre of the rotor shaft, see Figure 6. The suction gas then enters the compression chamber through another radial hole at the concentric and thus allowing the suction gas to flow radially to the suction chamber.

The above design concept forms the basic features of the RV compressor, the basic working principle of which will be described in the following section.

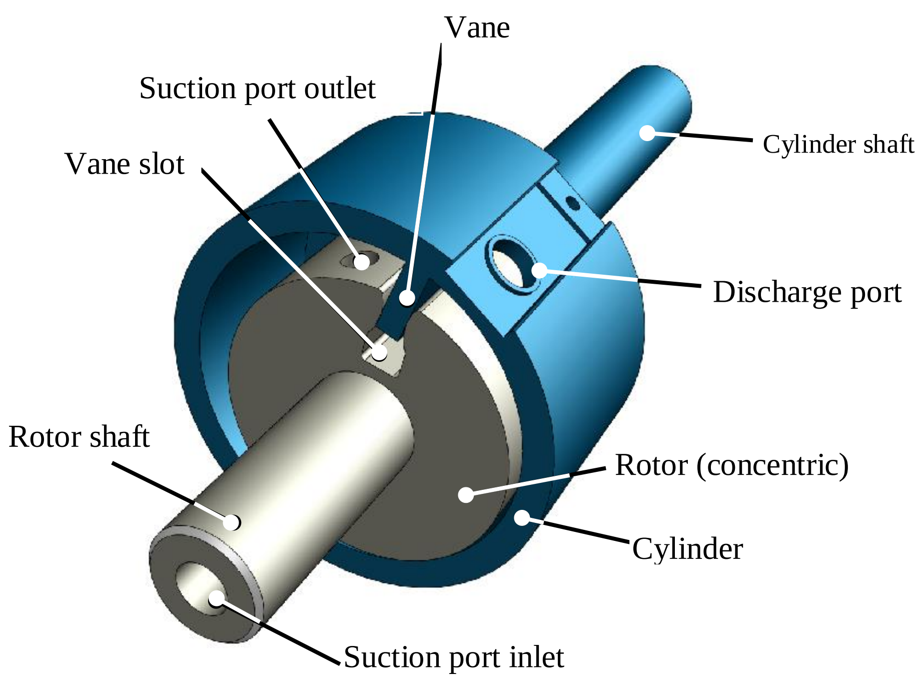



**Figure 6 A** Schematic of a Revolving Vane Compressor

## 3. The Revolving Vane Compressor (the RV)

### 3.1 Introduction to the RV

Figure 6 shows that in its basic form, the RV has three major components, namely a rotor or concentric (as opposed to the eccentric in the case of the RP), a vane (which is rigidly fixed to the cylinder) and a rotating cylinder (the shaft of which is connected to the motor). The suction and discharge ports are located at the concentric and the rotating cylinder, respectively. The RV does away with the roller and the vane spring in the RP. Figure 7 shows a sectional view of the RV. The centers of the cylinder and the rotor are offset such that there is a line contact at the outer circumference of the rotor and the inner wall of the cylinder, and this line contact separates the working chamber of the compressor into two: one suction and the other compression and discharge. The vane is rigidly attached to the cylinder and the other end of the vane slides inside the vane slot, the latter is located in the concentric. During the operation, the cylinder is directly driven by a motor, and the vane which is rigidly attached to the cylinder on one end, rotates the rotor at the other end. This causes the volume of the working chamber to vary in size and hence completes the working process of a compressor cycle.

It is noted that the rotating cylinder may aid the improvement of the volumetric efficiency when operating at high speeds by providing effects similar to that caused by the rotating impeller of a centrifugal pump. The centrifuge effects results in a higher gas density at the outer circumference of the working chamber and hence improves the volumetric efficiency of

the compressor. This 'centrifugal' effect is further enhanced by locating the suction port exit at a radial location in the concentric to facilitate the gas to flow radially outward into the suction chamber. For a more comprehensive analysis of the RV, readers may refer to refs [14-17].
In the RV, the rotor and the cylinder rotate about their own axis, the radial position of all parts remains the same during the operation, this situation allows a static balancing procedure to be sufficient in balancing the compressor dynamically.

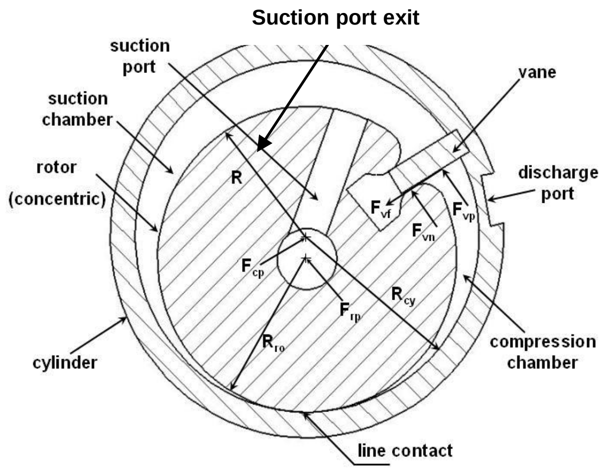



**Figure 7 A** Sectional View of a RV Compressor

### 3.2 Proof of Concept Study

Mathematical models [14-17] have been formulated to study the performance of the RV. These include the geometrical model, thermodynamics model, friction model, valve reed model and leakage model. Theoretical predictions show significant mechanical efficiency can be achieved [14]. The comparisons with that from other compressors are tabulated in Table 4. Figure 8 shows the typical predicted frictional loss in different parts of the RV operating at 3000 rev/min, with cooling capacity of 5.3kW when using R22 as the refrigerant.

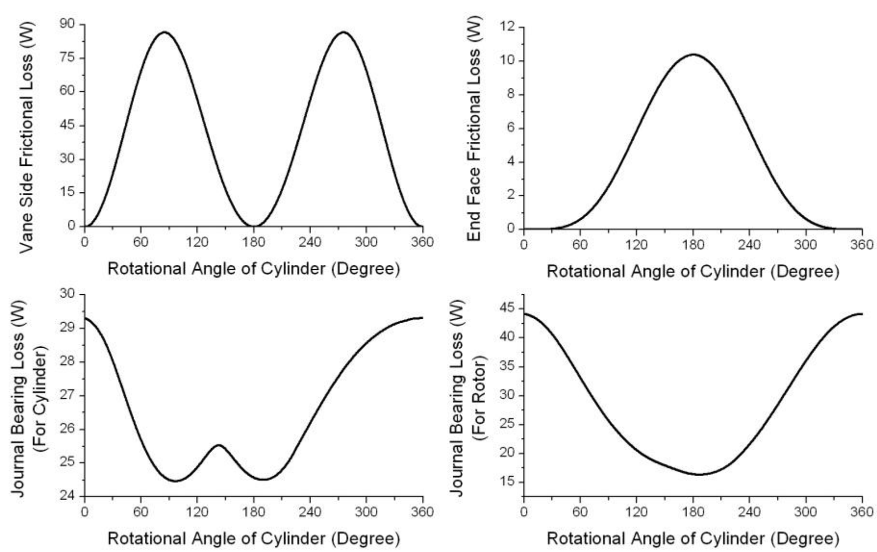



**Figure 8** Frictional Power Losses in the RV

**Table 4**: Comparison of Mechanical Efficiencies among Various Compressors

| S/N | Compressor Type | Mech. Efficiency (%) | Investigators |
|---|---|---|---|
| 1 | Rolling Piston | 85.0 – 90.0 | Ozu and Itami, 1981 [7] |
| 2 | Rolling Piston | 93.3 | Matsuzaka and Nagamoto, 1982 [8] |
| 3 | Rolling Piston | 92.5 | Wakabayashi et al., 1982 [9] |
| 4 | Rolling Piston | 92.0 -92.2 | Sakaino et al., 1984 [10] |
| 5 | Rolling Piston | 91.8 | Ishii et al., [11] |
| 6 | Scroll | 92.3 – 92.7 | Hayano et al., 1988 [18] |
| 7 | Scroll | 92.3 – 93.4 | Ishii et al., 1990 [19], 1992 [20], 1994 [21] |
| 8 | Revolving Vane | 96.5 | Teh and Ooi, 2008 [14] |

The proof of concept of operation of the initial version of the RV [15] has been carried out. Figure 9 shows the comparison between the measured and the predicted pressure variation in the compression chamber and Figure 10 shows the difference between measured and predicted shaft torque. The experiment was carried out using air as the working fluid and the RV compressor runs at 1000 rev/min. It is observed that there is good agreement between the theoretical and experimental results.

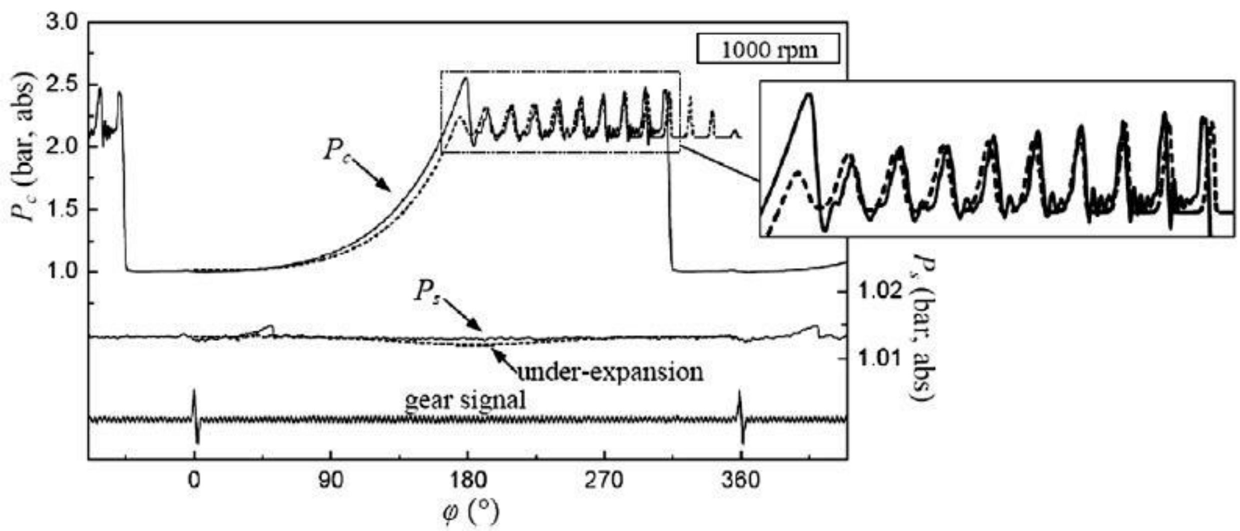



**Figure 9** Measured and Predicted Pressure Variation in the Compression Chamber [15]

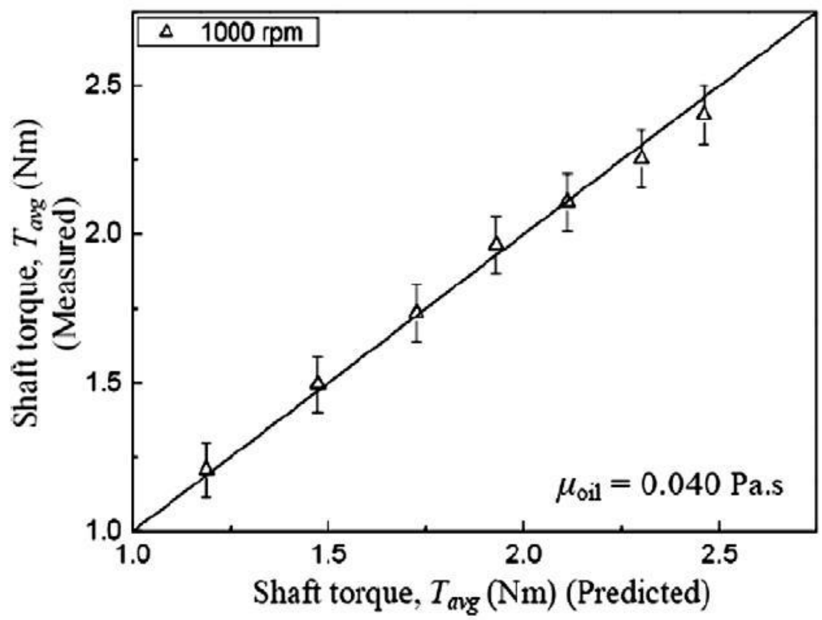



**Figure 10** Typical Comparisons between the Measured and Predicted Shaft Torque Required in Initial Prototype [15]

## 4. Concluding remarks

A step-by-step illustration showing the detailed design that aims at eliminating and reducing frictional losses in the rolling piston compressor leads to the inception of the new revolving vane compressor (the RV) was presented. The novel RV has the following potential advantages when compared to the rolling piston compressor:

1. It has three major components as opposed to five in the RP.
2. It has eliminated two frictional areas in the RP and these are at the vane-tip and at the inner surface of the roller.
3. It has significantly reduced frictional losses at all other areas, and these are at the vane slot, the endfaces of the cylinder and at the shaft bearings.
4. The vane side frictional loss is now independent of pressure differential across the vane in RV.
5. The shaft bearing load is significantly lower in RV as a result of the fat and short cylinder geometrical configuration.
6. The RV is geometrically simpler as all major parts are mostly circular and concentric, except the vane.
7. Once balance statically, it is balanced dynamically.

The initial evaluations show that revolving vane compressor is probably one of the best compressor designs currently available in terms of material cost, machining cost, reliability and performance. However, the evaluations must be verified by testing the compressor under the actual operating conditions in the compressor testing chamber. A descriptive comparison between the RP and the RV is also tabulated in Table 6.

**Table 6**: Comparison between rolling piston and revolving vane compressor

| Features | Rolling-Piston compressor | Revolving Vane compressor |
|---|---|---|
| 1. Geometry | More complicated component with an eccentric and roller | All components are concentric and no roller. |
| 2. Frictional loss | Higher frictional losses due to rubbing between components at:<br>i. roller and vane tip<br>ii. roller and eccentric<br>iii. roller and cylinder endfaces<br>iv. vane and slot<br>v. eccentric and cylinder endfaces<br>vi. journal bearings | Lower frictional losses due mainly to:<br>i. journal bearings |
| 3. Number of major component s | More major components, such as:<br>i. Cylinder<br>ii. Vane<br>iii. Shaft and eccentric<br>iv. Roller<br>v. Vane spring | Fewer major components, such as:<br>i. Cylinder<br>ii. Vane<br>iii. Shaft and concentric |
| 4. Precision and hardening processes | More critical components and require stringent precision and processes:<br>i. Vane – requires hardening to reduce wear at vane tip<br>ii. Roller - requires hardening to reduce wear<br>iii. Eccentric – eccentric offset from shaft requires precision control | No critical components |
| 5. Costs | i. Higher material costs – due to more components<br>ii. Higher process costs – due to more critical components<br>iii. Higher motor costs – a bigger motor is needed due to:<br>a. more frictional loss<br>b. higher max torque | i. Lower material costs – due to fewer components<br>ii. Lower process costs – due to :<br>a. no critical components<br>b. no eccentric<br>iii. Lower motor costs – a smaller motor is needed due to:<br>c. less frictional loss<br>d. lower max torque |
| 6. Efficiency | Lower efficiency due to:<br>i. higher frictional losses<br>ii. higher internal leakage due to:<br>a. larger average operating radial clearance between roller and cylinder.<br>b. larger allowable axial clearance at cylinder | Higher efficiency due to:<br>i. lower frictional losses<br>ii. lower internal leakage due to:<br>a. smaller average operating radial clearance between concentric and cylinder<br>b. smaller allowable axial |

| | | |
|---|---|---|
| | endfaces. | clearance at cylinder endfaces<br>ii. centrifugal compression due to rotating cylinder and ports' location<br>iii. lower discharge loss due to centrifugal softening of valve reed |
| 7. Reliability | Lower reliability due to:<br>i. more critical components<br>ii. more wear | Higher reliability due to:<br>i. no critical components<br>ii. less wear |

**References**


[1] JARN, Japan Air conditioning, heating & Refrigeration news, Serial no. 469-S, Special Edition, February 25, 2008

[2] K. T. Ooi, T. N. Wong, A computer simulation of a rotary compressor for household refrigerators, Applied Thermal Engineering, Volume 17, Issue 1, January 1997, Pages 65-78,

[3] Ooi, K.T., 2005, Design optimization of a rolling piston compressor for refrigerators, *Applied Thermal Engineering*, 25: 813–829

[4] Yanagisawa, T., Shimizu T., 1985, Leakage losses with a rolling piston type rotary compressor. I. Radial clearance on the rolling piston, *Int. J. of Ref.*, vol. 8, no. 2:p75-84

[5] Yanagisawa, T., Shimizu T., 1985, Leakage losses with a rolling piston type rotary compressor. II. Leakage losses through clearances on rolling piston faces, *Int. J. of Ref.*, vol. 8, no. 3:p152-158

[6] Yanagisawa, T., Shimizu T., 1985, Friction Losses in Rolling Piston Type Rotary Compressor. III, *Int. J. Refrig*., vol. 8, no. 3: p159-165

[7] Ozu, M., Itami, T., 1981, Efficiency Analysis of Power Consumption in Small Hermetic Refrigerant Rotary Compressors, International Journal of Refrigeration, 4(5), 265-270

[8] Matsuzaka, T., Nagatomo, S., 1982, Rolling Piston Type Rotary Compressor Performance Analysis, Proc. Purdue Compressor Technology Conference, 149-158

[9] Wakabayashi, H., Yuuda, J., Aizawa, T., Yamamura, M., 1982, Analysis of Performance in a Rotary Compressor, Purdue Compressor Technology Conference, 140-147

[10] Sakaino, K., Muramatsu, S., Shida, S., Ohinata, O., 1984, Some Approaches towards a High Efficient Rotary compressor, Proc. Purdue Compressor Technology Conference, 315-322

[11] Ishii, N., Fukushima, M., Yamamura, M., Fujiwara, S., Kakita, S., 1990, Optimum Combination of Parameters for High Mechanical Efficiency of a Rolling-Piston Rotary Compressor, Proc. Purdue Compressor Technology Conference, 18-424

[12] Zogg, M, History of heat pumps Swiss contributions and international milestones, 9th International IEA Heat Pump Conference, 20 – 22 May 2008, Zürich, Switzerland

[13] K.T. Ooi, Assessment of a rotary compressor performance operating at transcritical carbon dioxide cycles, Applied Thermal Engineering, Volume 28, Issue 10, July 2008, Pages 1160-1167, ISSN 1359-4311

[14] Y.L. Teh, K.T. Ooi, Theoretical study of a novel refrigeration compressor - Part I: Design of the revolving vane (RV) compressor and its frictional losses, International Journal of Refrigeration, Volume 32, Issue 5, August 2009, Pages 1092-1102

[15] Y.L. Teh, K.T. Ooi, D. Wibowo Djamari, Theoretical study of a novel refrigeration compressor - Part II: Performance of a rotating discharge valve in the revolving vane (RV) compressor, International Journal of Refrigeration, Volume 32, Issue 5, August 2009, Pages 1103-1111
[16] Y.L. Teh, K.T. Ooi, Theoretical study of a novel refrigeration compressor- Part III: Leakage loss of the revolving vane (RV) compressor and a comparison with that of the rolling piston type, International Journal of Refrigeration, Volume 32, Issue 5, August 2009, Pages 945-952
[17] Y.L. Teh, K.T. Ooi, Experimental study of the Revolving Vane (RV) compressor, Applied Thermal Engineering, Volume 29, Issues 14-15, October 2009, Pages 3235-3245
[18] Hayano, M., Sakata, H., Nagatomo, S., Murasaki, H., 1988, An Analysis of Losses in Scroll Compressor, Proc. Purdue Compressor Technology Conference, 189-197
[19] Ishii, N., Yamamura, M., Muramatsu, S., Yamamoto, S., Sakai, M., 1990, Mechanical Efficiency of a Variable Speed Scroll Compressor, Proc. Purdue Compressor Technology Conference, 192-199
[20] Ishii, N., Yamamoto, S., Muramatsu, S., Yamamura, M., Takahashi, M., 1992, Optimum Combination of Parameters for High Mechanical Efficiency of a Scroll Compressor, Proc. Purdue Compressor Technology Conference, 118a1-118a8
[21] Ishii, N., Yamamura, M., Muramatsu, S., Yamada, S., Takahashi, M., 1994, A Study on High Mechanical Efficiency of a Scroll Compressor with Fixed Cylinder Diameter, Proc. Purdue Compressor Technology Conference, 677-682