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

Teh, Yong Liang; Ooi, Kim Tiow

2008

Teh, Y. L., & Ooi, K. T. (2008). 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*, 1-29.

<https://hdl.handle.net/10356/79785>

<https://doi.org/10.1016/j.ijrefrig.2008.11.001>

Copyright © 2009 Elsevier Ltd and the International Institute of Refrigeration (IIR). Journal homepage: www.elsevier.com/locate/ijrefrig

Downloaded on 26 Apr 2023 17:48:03 SGT

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

Y. L. Teh¹ (Ph.D. Student, corresponding author), K. T. Ooi² (Associate Professor)

Thermal & Fluids Engineering Division, School of Mechanical & Aerospace Engineering, Nanyang Technological University, 50 Nanyang Avenue, S639798, Singapore

¹ Email: tehy0002@ntu.edu.sg, Tel: +65 6790 6174, Fax: +65 6792 4062

² Email: mktooi@ntu.edu.sg, Tel: +65 6790 5511, Fax: +65 6792 4062

Abstract

A new refrigeration compressor, named ‘Revolving Vane (RV) compressor’, has been introduced in Part I of this paper series. The efficiency of the RV compressor, alike other refrigeration compressors, is primarily affected by mechanical and leakage losses, amongst which the former has been investigated in Part I. In the present paper, the leakage loss in the RV compressor which mainly occurs at the radial clearance between the rotor and the cylinder is analyzed theoretically. In comparison with the rolling piston type, the leakage loss at the radial clearance in the RV compressor is typically found to be 40 % lesser than that of the former. Furthermore, the leakage loss can be significantly reduced by selecting a shorter configuration of the compressor, which is anticipated to yield good volumetric efficiencies of more than 95 %. For a longer configuration, the leakage loss can be effectively minimized by employing the method of an eccentric cylinder.

Keywords

Leakage, Modelling, Loss, Volumetric efficiency, Comparison, Rotary compressor

Nomenclature

e_{rc}	distance between rotor and cylinder journal centers [m]
F	force [N]
L	length [m]
L_c	axial length of working volume [m]

L_f	friction length of leakage path [m]
P	pressure [Pa]
q_m	leakage mass flow rate [kg s ⁻¹]
R	prescribed radius [m]
R_g	gas constant [J kg ⁻¹ K ⁻¹]
r_v	radial distance from rotor center to vane tip center [m]
T	temperature [K]
t	time [s]
V	velocity [m s ⁻¹]
β	bearing force angle [rad]
δ	dynamic radial clearance [m]
δ_0	assembly radial clearance [m]
δ_b	prescribed radial clearance of bearing [m]
ε	bearing eccentricity ratio [-]
η_l	leakage loss (decrease in volumetric efficiency) = $q_m/\text{ideal suction mass flow rate} \times 100$ [%]
η_{vs}	kinetic friction coefficient at vane sides [-]
μ	dynamic viscosity of lubricant [Pa s]
φ	driver/rotor angle [rad]
φ_{cy}	cylinder angle [rad]
Φ	bearing attitude angle [rad]
ω	angular velocity [rad s ⁻¹]
Subscripts	
b	of bearing
c	compression
cy	of cylinder
d	discharge
e	at model channel exit
I	inertia
ro	of rotor
s	suction

1. Introduction

1.1 Leakage loss in a compressor

Leakage loss at the compressor is one of the most important factors that affect the performance of a refrigeration cycle. In rotary compressors widely used for air-conditioning, such as the rolling piston and the scroll types, leakage losses arise due to the existence of clearances required for the smooth and reliable operation of the compressor mechanism. For example, during the operation of the rolling piston compressor, it is found that internal leakages of the refrigerant gas occur at the radial and endface clearances of the rolling piston (Chu et al., 1978, Pandeya and Soedel, 1978, Ozu and Itami, 1981, Yanagisawa and Shimizu, 1985a, 1985b). Similarly in scroll compressors, there also exist axial and radial clearances between the meshing scrolls at which leakage flows occur (Ishii 1996a). Generally, such leakage losses are limited by controlling the tolerances of attributive components during manufacturing, which at times may be impractical as a high level of precision is required. Therefore, a certain degree of leakage loss in the compressor is usually tolerated which brings about a lower level of efficiency in the refrigeration system.

Although leakage losses are somewhat unavoidable, they can however be effectively minimized by introducing a compressor design with coherent design features. In such a design, the inherent characteristic of the compressor is such that when one aspect of the compressor is improved, it would not cause negative effects to other aspects. This is usually not the case with today's compressor designs. For example, in the rolling piston compressor, it has been found that leakage loss at the endface clearance of the rolling piston is highly sensitive to the size of the clearance (Yanagisawa and Shimizu, 1985b). When the clearance is decreased to reduce the leakage, the frictional loss due to viscous shear of the oil film between the endfaces of the rolling piston and the stationary cylinder can be expected to increase. In such a situation, the volumetric efficiency is improved at the expense of the mechanical efficiency. One can agree that a similar circumstance is also present for the scroll compressor at the axial clearances between the scroll tips and the base plate. In such dire straits the overall efficiency of the compressor will be limited. Hence, a first feature of a coherent compressor design enables the leakage loss to be readily reduced without compromising other performance aspects of the compressor.

The second of such features relates to reliability considerations. For example, in the rolling piston compressor, it has been reported that the leakage loss at the radial clearance of the rolling piston is

aggravated due to a dramatic increase of the radial clearance during the later part of the working cycle as result of the journal motion of the bearing (Yanagisawa and Shimizu, 1985a). When one attempts to decrease the radial clearance during assembly to reduce the leakage, there is a limit to the decrement beyond which collision of the rolling piston with the cylinder may occur at some point during operation. In such a situation, the improvement of the volumetric efficiency is constrained by reliability requirements. On the other hand, in a coherent compressor design, the reduction of the leakage can be achieved without compromising the reliability of the machine.

Last but not least, the third coherent design feature relates to the geometrical adjustment of the basic dimensions of a compressor to minimize its leakage losses. Usually, in a particular compressor type, there exist certain geometrical configurations which favors the volumetric efficiency, and others the mechanical efficiency. For example, it is shown by Ishii et al. (1996b) that the respective maximum volumetric and mechanical efficiencies of the scroll compressor are attained at substantially different radiuses of the involute base circle. Thus, this creates a dilemmatic situation which limits the highest overall efficiency of the compressor. However, if the compressor is designed with coherent features, the various geometrical configurations converge to one common optimum in which leakage and mechanical losses are simultaneously minimized.

The above discussion briefly highlights the important factors to be considered when reducing the leakage loss in a compressor through various means. In this paper, we theoretically analyze the leakage loss of a new compressor, named ‘Revolving Vane (RV) Compressor’, and conduct a comparison study with the rolling piston type. In addition, a parametric study is performed to elucidate a configuration of the RV compressor with the highest volumetric efficiency.

1.2 Leakage loss in the RV compressor

Fig. 1 shows the basic construction of the RV compressor (Teh and Ooi, *Submitted* 2008) which mainly comprises of a rotor, a vane and a cylinder. The rotor and the cylinder are assembled with an eccentricity such that a virtual line contact exists between the two components (at $\varphi = 0$ rad). Both the rotor and cylinder are supported individually and concentrically on bearing pairs and allowed to rotate about their respective axes of rotation. During operation, the rotation of the rotor revolves the vane which in turn rotates the cylinder. The motion causes the volumes trapped within the rotor, vane and cylinder to vary, resulting in suction, compression and discharge of the working fluid. Due to the rotation of the

entire cylinder on which the discharge valve is being mounted, the valve reed experiences centrifugal forces in addition to pressure forces. A detailed dynamic analysis of the rotating valve and its effect on the compression efficiency is addressed in Part II of this paper series (Teh et al., *Submitted* 2008).

Similar to the rolling piston compressor, internal leakages of the refrigerant gas can occur in the RV compressor at the radial and endface clearances between the rotor and the cylinder. However, as the rotor and cylinder are both rotating, the relative velocities between opposite surfaces at each endface clearance are small. This allows the reduction of the axial clearance at each endface without noticeably increasing the endface friction (Teh and Ooi, *Submitted* 2008), but however readily reduces the leakage loss at that region. According to Yanagisawa and Shimizu (1985b), when the endface clearance in the rolling piston compressor is below 10 μm , the decrease of volumetric efficiency due to the endface leakage becomes negligible. Thus, by enforcing such a condition in the RV compressor, the endface leakage can be eliminated and the only significant source of internal leakage is that left at the radial clearance.

2. Theoretical Model

2.1 Radial clearance between rotor and cylinder

For smooth operation of the new compressor mechanism, a radial clearance gap exists at the virtual line contact between the external surface of the rotor and the interior surface of the cylinder. The working fluid therefore leaks from the compression chamber to the suction chamber through the radial clearance due to a pressure difference. During assembly, the radial clearance is statically defined by the rotor and cylinder radiuses and the distance between the rotor and cylinder bearing centers. However, during operation, the positions of the rotor and cylinder journals change dynamically which causes variations in the clearance gap. As the amount of leakage is very sensitive to the cross-sectional area of the flow, any change in the radial clearance affects the leakage loss to a large extent. It is thus important to understand the dynamic characteristics of the radial clearance during operation and find means of minimizing the clearance gap.

The dynamic radial clearance, δ , and its attributive components are illustrated in Fig. 2. Due to the motions of the rotor and cylinder journals, δ varies according to the equation:

$$\delta = \delta_0 + (R_{cy} - R_{ro}) - e_{rc} \quad (1)$$

in which δ_0 is the assembly radial clearance when both the rotor and cylinder journals are respectively concentric to their bearing centers. The distance between the journal centers has the geometrical relation:

$$e_{rc} = \sqrt{e_{rc,x}^2 + e_{rc,y}^2} \quad (2)$$

where the orthogonal components can be found to be respectively defined by:

$$\begin{aligned} e_{rc,x} &= \delta_{b,ro} \varepsilon_{ro} \sin(\Phi_{ro} + \beta_{ro}) - \delta_{b,cy} \varepsilon_{cy} \sin(\Phi_{cy} + \beta_{cy}) \\ e_{rc,y} &= \delta_{b,ro} \varepsilon_{ro} \cos(\Phi_{ro} + \beta_{ro}) - \delta_{b,cy} \varepsilon_{cy} \cos(\Phi_{cy} + \beta_{cy}) + R_{cy} - R_{ro} \end{aligned} \quad (3)$$

Through Eqs. (1) to (3), it is clear that the radial clearance changes according to the relative positions of the rotor and cylinder journal centers, defined by their respective eccentricities and attitude angles. In this investigation, the characteristics of the journal bearings are calculated based on a method of solution for dynamically loaded finite length bearings proposed by Hirani et al. (1999). The radial and angular velocities of the journal centers are respectively given by:

$$\dot{\varepsilon}_J = \frac{\sqrt{F_{bx,J}^2 + F_{by,J}^2} (\delta_{b,J} / R_{b,J})^2}{6\mu L_{b,J} R_{b,J}} M_J^\varepsilon, \quad \dot{\Phi}_J = \frac{\sqrt{F_{bx,J}^2 + F_{by,J}^2} (\delta_{b,J} / R_{b,J})^2}{6\mu L_{b,J} R_{b,J} \varepsilon_J} M_J^\Phi + \frac{\omega_J}{2} - \dot{\beta}_J \quad (4)$$

where subscript $J = ro, cy$ for the respective rotor and cylinder journals. M_J^ε and M_J^Φ are mobility components defined in the reference. The resultant force on the rotor journal is caused by the pressure forces and the contact forces at the vane sides, whereas for the cylinder journal it is due to the pressure forces, the vane tip force and the inertia force of the cylinder (Teh and Ooi, *Submitted* 2008), i.e.

$$\begin{aligned} F_{bx,ro} &= (F_2 - F_1) \cos \varphi + F_{sx,ro} - F_{cx,ro} - \eta_{vs} (|F_1| + |F_2|) \operatorname{sgn}(\dot{r}_v) \sin \varphi, \\ F_{by,ro} &= (F_1 - F_2) \sin \varphi + F_{sy,ro} - F_{cy,ro} - \eta_{vs} (|F_1| + |F_2|) \operatorname{sgn}(\dot{r}_v) \cos \varphi \end{aligned} \quad (5)$$

$$\begin{aligned} F_{bx,cy} &= -F_{vt,r} \sin \varphi - F_{I,cy} \cos \varphi_{cy} - F_{sx,cy} + F_{cx,cy}, \\ F_{by,cy} &= -F_{vt,r} \cos \varphi + F_{I,cy} \sin \varphi_{cy} - F_{sy,cy} + F_{cy,cy} \end{aligned} \quad (6)$$

It is to be noted that although the rotor journal rotates at a constant velocity, the angular velocity of the cylinder journal is constantly varying due to the offset between the bearing centers, such that it follows the expression:

$$\omega_{cy} = \frac{d\varphi}{dt} \frac{d\varphi_{cy}}{d\varphi} = \omega_{ro} \frac{d}{d\varphi} \left\{ \cos^{-1} \left[\frac{R_{cy}^2 + (R_{cy} - R_{ro})^2 - r_v^2}{2R_{cy}(R_{cy} - R_{ro})} \right] \right\} \quad (7)$$

For simplicity of the analysis, it is assumed that the upper and lower bearings are perfectly aligned and have identical dimensions, for both the rotor and the cylinder, such that each respective journal load is divided equally between the upper and lower bearings.

2.2 Leakage mass flow rate across rotor radial clearance

Leakage of the refrigerant gas occurs from the compression chamber to the suction chamber because of the existence of the clearance gap. As the geometry of the leakage channel in the RV compressor is exactly similar to that in the rolling piston compressor, the leakage flow across the rotor radial clearance in the RV compressor is modelled using a method proposed by Yanagisawa and Shimizu (1985a). The method takes into account of the viscous friction imposed on the flow due to the leakage path being long and narrow as compared to its height, which has achieved good agreement between theoretical predictions and experimental results. Mainly, the leakage path across the radial clearance between the rotor and the cylinder (Fig. 3a) is modelled using the flow channel shown in Fig. 3b, which consists of a compression chamber, a convergent nozzle, a straight channel imposing viscous drag on the flow and a suction chamber. The friction length L_f is found by equating the frictional loss in the model channel to that developed in the actual leakage path, which can be found to give the expression:

$$L_f = \frac{2\pi\delta R_{cy}}{(R_{cy} - R_{ro})\sqrt{1 - (R_{cy} - R_{ro})^2}} \quad (8)$$

Subsequently, by considering isentropic flow across the convergent nozzle and adiabatic frictional (Fanno) flow across the straight channel, the pressure, P_e , temperature, T_e and flow velocity, V_e , at the channel exit can be found by comparing the pressure ratio across the entire model leakage path to the known pressure ratio between the compression and suction chambers. The leakage mass flow rate is then calculated by the relation:

$$q_m = \delta L_c V_e \left(\frac{P_e}{R_g T_e} \right) \quad (9)$$

3. Results and Comparison

For the purpose of comparison with the rolling piston compressor (Yanagisawa and Shimizu, 1985a), the operating specifications and main dimensions of the RV compressor being investigated are similar to that of the former, as shown in Table 1. Fig. 4a shows the variation of the radial clearance, δ , and the distance between the rotor and cylinder journal centers, e_{rc} , for one complete shaft revolution. Generally, δ changes in a similar manner to that in the rolling piston compressor (δ_{RP}), and increases above its assembly value, δ_0 , for most part of the cycle. This is due to the motion of the rotor and cylinder journals under the dynamic loads, which causes e_{rc} to decrease below its assembly value, $R_{cy} - R_{ro}$, especially in the vicinity of $\varphi = 0$ rads. In order to better understand the influence of the journal motions on the radial clearance, the respective bearing loads, $F_{b,ro}$ and $F_{b,cy}$, and loci, $\mathcal{E}_{ro}$ and $\mathcal{E}_{cy}$, are plotted in polar coordinates as shown in Fig. 5a. It is observed that the loci of the rotor and cylinder journals are highly symmetrical to each other, due to the respective bearing forces largely remaining in opposite directions with equivalent magnitudes throughout the working cycle. For both the rotor and cylinder bearings, the eccentricities decrease in the region of $\varphi = 0$ to $\pi +$ rads, and increase from there onwards to the end of the cycle, even though the bearing loads decrease in the later half of the cycle. This is because the RV compressor, alike the rolling piston compressor, has bearing forces that change their direction by half the speed of the shaft rotation. This causes the eccentricities to ‘overshoot’ resulting in a decrease in the load capacity of each journal bearing.

Although the radial clearance for the RV compressor varies similarly to that in the rolling piston compressor, it is however lesser for the former, which thus alleviates the leakage flow in comparison to that in the rolling piston design. Fig. 4b shows the variation of the instantaneous leakage mass flow rate in both compressors when they are subjected to similar pressure changes. As compared to the rolling piston compressor, it is clearly observed that the leakage flow across the radial clearance is less severe in the RV compressor, especially during the second half of the cycle when the pressure difference across the leakage path is large. There are two reasons for the improvement in the new compressor, which can be explained by comparing Figs. 5a and 5b. Firstly, unlike the rolling piston compressor, there is no piston bearing in the RV compressor which has been found to exhibit a high eccentricity, especially towards the end of the shaft revolution. The high eccentricity causes the radial clearance to be increased which results in more

leakage flow and is further aggravated by the large pressure difference. Secondly, although both compressors experience similar pressure changes in their working chambers, the design of the RV compressor exhibits lesser bearing loads, with a lower maximum of about 2.0 kN as compared to 2.2 kN in the rolling piston design. This is because unlike the rolling piston compressor, the bearings in the RV compressor do not experience a large vane tip contact force. The lower bearing forces results in smaller bearing eccentricities, contributing to the reduced radial clearance. Thus, with the same dimensions, a lower leakage loss at the radial clearance is clearly observed in the RV compressor as compared to the rolling piston design.

To this end, it is important to realize that in Fig. 4a, the minimum radial clearance, $\delta_{\min}$, which occur in the vicinity of π radians for both compressors, differ. This is because due to the reasons mentioned above, the variation amplitude of δ in the RV compressor is smaller. Usually, a minimum radial clearance is provided throughout the entire shaft revolution to ensure a safe operation of the compressor such that no collision between the rotor/rolling piston and the cylinder can occur. In this aspect of ensuring operational reliability, δ_0 of the RV compressor can be decreased from 20 μm to 15 μm such that $\delta_{\min}$ for both compressors becomes the same, at about 14 μm . When this happens, the leakage mass flow rate of the RV compressor is decreased further as shown in Fig. 4b. Effectively, the leakage loss at the radial clearance is reduced by more than 40 % in the RV compressor as compared to that in the rolling piston design.

4. Parametric Analysis

In the following sections, we investigate the effects of some parameters on the internal leakage at the rotor radial clearance, with the objective of elucidating a configuration that maximizes the volumetric efficiency of the new design.

4.1 Effect of rotor-to-cylinder radius ratio

The basic geometrical configuration of the RV compressor is mainly dictated by the radiuses of the rotor and the cylinder, R_{ro} and R_{cy} , and the axial length, L_c . Here, we investigate the effects of varying the rotor-to-cylinder radius ratio, R_{ro}/R_{cy} , over a practical range by changing R_{cy} and L_c ,

while maintaining the volumetric capacity and other parameters specified in Table 1. Fig. 6a shows an exponential increase in the leakage loss, η_l , when R_{ro}/R_{cy} is increased from 0.75 to 0.95. This is mainly due to the increase in L_c when the rotor-to-cylinder radius ratio increases, which results in a larger width of the leakage path. Furthermore, as the compressor lengthens, the pressure difference between the suction and compression chambers has more area to act on which enlarges the bearing loads, resulting in bigger eccentricities of the journals. In consequence, δ increases as shown in Fig. 6b. The widening of both dimensions, L_c and δ , inevitably allows the leakage flow to increase drastically. As such, a smaller R_{ro}/R_{cy} thus L_c/R_{cy} , which leads to a shorter configuration of the RV compressor, is preferred for lesser leakage losses. However, this preference is in contrary to minimizing the frictional losses, where it is shown that a longer configuration exhibits lesser friction in Part I of this paper series (Teh and Ooi, *Submitted* 2008). But, it is found that the large frictional loss at small values of R_{ro}/R_{cy} is mainly contributed by the vane side friction. Thus, it is essential to find means of effectively reducing the vane side friction in order to simultaneously achieve maximum mechanical and volumetric efficiencies in the RV compressor.

4.2 Effect of eccentric cylinder with reduction of rotor radial clearance

In the rolling piston compressor, Yanagisawa and Shimizu (1985a) showed that a decrease in δ_0 coupled with an eccentric assembly of the main bearing can considerably reduce the leakage loss at the radial clearance. In this method, the leakage flow rate is effectively reduced by maintaining the radial clearance in the proximity of a minimum and safe value throughout the entire working cycle, especially during the second half of the shaft revolution during which the pressure difference across the leakage gap is large. Due to similar geometrical characteristics, such a method can also be applied to the RV compressor. However, in the RV design, as the rotor and cylinder are both concentric to their respective journal centers, an eccentric assembly of the bearings would not result in the same effect as that demonstrated in the rolling piston compressor. Instead, for the RV compressor, the cylinder is fabricated with an eccentricity, δ_e , in the direction, φ_e , relative to its journal center, as illustrated in Fig. 7. In this case, for the calculation of δ , the components of e_{rc} originally defined in Eq. (3) are to be replaced by:

$$\begin{aligned}
e_{rc,x} &= \delta_e \sin(\varphi_e + \varphi_{cy}) + \delta_{b,ro} \varepsilon_{ro} \sin(\Phi_{ro} + \beta_{ro}) - \delta_{b,cy} \varepsilon_{cy} \sin(\Phi_{cy} + \beta_{cy}) \\
e_{rc,y} &= -\delta_e \cos(\varphi_e + \varphi_{cy}) + \delta_{b,ro} \varepsilon_{ro} \cos(\Phi_{ro} + \beta_{ro}) - \delta_{b,cy} \varepsilon_{cy} \cos(\Phi_{cy} + \beta_{cy}) + R_{cy} - R_{ro}
\end{aligned} \tag{10}$$

Fig. 8a shows the variation of δ when the cylinder is fabricated with $\delta_e = 6 \mu\text{m}$ at various angles of φ_e . Expectedly, as the original variation of δ shown in Fig. 4a somewhat follows a cosine function, its amplitude of variation can be decreased by incorporating an appropriate value of δ_e in the direction of $\varphi_e = \pi$ rads. Here, the value of δ_e is chosen such that it minimizes the variation amplitude of δ with a smaller magnitude occurring in the later part of the working cycle. Then, the assembly clearance, δ_0 , is reduced to lower $\delta_{\min}$. As shown in Fig. 8a, at $\delta_0 = 10 \mu\text{m}$, δ is maintained in the vicinity of and above a $\delta_{\min}$ of $14 \mu\text{m}$ throughout the working cycle, which is the same condition as that imposed earlier. However, Fig. 8b shows that with the eccentric cylinder, the leakage loss is further decreased by another 2 to 3 percent over the entire practical range of δ_0 .

4.3 Effect of eccentric cylinder with reduction of rotor radial clearance at different rotor-to-cylinder radius ratios

In this section, the above method of an eccentric cylinder is applied to various geometrical configurations of the RV compressor having different values of R_{ro}/R_{cy} . Fig. 9 shows a comparison of the leakage losses between the use of an eccentric cylinder and a concentric cylinder, where in both cases the radial clearance is maintained above a $\delta_{\min}$ of $14 \mu\text{m}$. The eccentric cylinder method is clearly observed to be more effective at higher ratios of R_{ro}/R_{cy} when the RV compressor is having a longer cylinder. This is because in the use of the concentric cylinder, the variation of δ enlarges when R_{ro}/R_{cy} increases as shown in Fig. 6b, which limits the direct reduction of δ_0 to decrease the leakage loss without infringing the value of $\delta_{\min}$ to be maintained. Therefore, it is only necessary to employ an eccentric cylinder, which introduces additional precision requirements during fabrication, at longer configurations of the RV compressor.

5. Conclusion

In this paper, the leakage loss of the RV compressor which mainly occurs at the radial clearance between the rotor and the cylinder is theoretically analyzed. As compared to the rolling piston compressor, a reduction of more than 40 % in the leakage loss at the radial clearance has been predicted in the new design. Furthermore, in effort to maximize the overall efficiency of the RV compressor, a parametric study has been conducted which resulted in the following conclusions:

- A shorter configuration of the RV compressor is anticipated to attain higher volumetric efficiencies of more than 95 % but lower mechanical efficiencies. It is thus important to reduce the vane side friction in order to also achieve a higher mechanical efficiency of the compressor when this configuration is used.
- The method of an eccentric cylinder, which reduces the leakage loss, is more effective when the RV compressor is of a longer configuration. Thus, the eccentric cylinder can be employed in a longer configuration of the compressor, which has a higher mechanical efficiency.
- An improved design of the RV compressor which eliminates the vane side friction should be explored. If achieved, it can be expected to attain unprecedented COP values of a refrigeration compressor.

In this paper series, theoretical studies presented in Part I, II and III has shown good potential of the RV compressor to attain significant improvement in energy efficiency. Hence, practical development has since commenced. Experimental verification of the actual performance of the new compressor will be reported in a future paper.

References

- Chu, I., Shiga, T., Ishijima, K., Sakaino, M., 1978, Analysis of the Rolling-Piston Type Rotary Compressor, Proc. Purdue Compressor Technology Conference, 219-225
- Hirani, H., Athre, H., Biswas, S., 1999, Dynamically Loaded Finite Length Journal Bearings: Analytical Method of Solution, ASME Journal of Tribology, 121, 844-852
- Ishii, N., Bird, K., Sano, K., Oono, M., Iwamura, S., Otokura, T., 1996a, Refrigerant Leakage Flow Evaluation for Scroll Compressors, Proc. Purdue Compressor Technology Conference, 633-638

- 1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
19
20
21
22
23
24
25
26
27
28
29
30
31
32
33
34
35
36
37
38
39
40
41
42
43
44
45
46
47
48
49
50
51
52
53
54
55
56
57
58
59
60
61
62
63
64
65
- Ishii, N., Sakai, M., Sano, K., Yamamoto, S., Otokura T., 1996b, A Fundamental Optimum Design for High Mechanical and Volumetric Efficiency of Compact Scroll Compressors, Proc. Purdue Compressor Technology Conference, 639-644
- Ozu, M., Itami, T., 1981, Efficiency Analysis of Power Consumption in Small Hermetic Refrigerant Rotary Compressors, International Journal of Refrigeration, 4(5), 265-270
- Pandeya, P., Soedel, W., 1978, Rolling Piston Type Compressors with Special Attention to Friction and Leakage, Proc. Purdue Compressor Technology Conference, 209-218
- Teh, Y. L., Ooi, K. T., Theoretical Study of a Novel Refrigeration Compressor - Part I: Design of the Revolving Vane (RV) Compressor and its Frictional Losses, International Journal of Refrigeration, *Submitted* 2008
- Teh, Y. L., Ooi, K. T., Wibowo Djamari, D., 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, *Submitted* 2008
- Yanagisawa, T., Shimizu, T., 1985a, Leakage Losses with a Rolling Piston Type Rotary Compressor. I. Radial Clearance on the Rolling Piston, International Journal of Refrigeration, 8(2), 75-84
- Yanagisawa, T., Shimizu, T., 1985b, Leakage Losses with a Rolling Piston Type Rotary Compressor. II. Leakage Losses through Clearance on Rolling Piston Faces, International Journal of Refrigeration, 8(3), 152-158

Figures

Fig. 1 – Schematic sections of the RV compressor: (a) End view, (b) Side view.

Fig. 2 – Definition of the dynamic radial clearance, δ , and its attributive components (Note: Eccentricities are exaggerated for clarity – the position of the virtual line contact remains in the vicinity of $\varphi = 0$ rad.).

Fig. 3 – Modelling of leakage path: (a) Actual flow path, (b) Model flow path.

Fig. 4 – (a) Variation of radial clearance and distance between journal centers (Note: δ_0 is the same for both compressors.); (b) Variation of instantaneous leakage mass flow rate at radial clearance.

Fig. 5 – Variation of bearing load and loci of journal centers for: (a) RV compressor, (b) rolling piston compressor.

Fig. 6 – Effect of rotor-to-cylinder radius ratio on: (a) leakage loss and compressor configuration ($R_{ro} = 23.8$ mm), (b) instantaneous leakage mass flow rate and dynamic radial clearance.

Fig. 7 – Definition of eccentric cylinder (Note: Rotor journal bearing is not shown for clarity.)

Fig. 8 – (a) Variation of radial clearance for an eccentric cylinder ($\delta_e = 6$ μ m) at various φ_e ; (b) Comparison of leakage loss between concentric and eccentric cylinders.

Fig. 9 – Effect of eccentric cylinder on leakage loss at various rotor-to-cylinder radius ratios ($\delta_{\min} = 14$ μ m).

Figure 1a

[Click here to download high resolution image](#)

Figure 1b
[Click here to download high resolution image](#)

Figure 2
[Click here to download high resolution image](#)

Figure 3
[Click here to download high resolution image](#)

Figure 4a
[Click here to download high resolution image](#)

(a)

Figure 4b
[Click here to download high resolution image](#)

(b)

Figure 5a
[Click here to download high resolution image](#)

Figure 5b
[Click here to download high resolution image](#)

(b)

Figure 6a
[Click here to download high resolution image](#)

(a)

Figure 6b
[Click here to download high resolution image](#)

(b)

Figure 7
[Click here to download high resolution image](#)

Figure 8a
[Click here to download high resolution image](#)

(a)

Figure 8b
[Click here to download high resolution image](#)

(b)

Figure 9
[Click here to download high resolution image](#)

Table 1 – Operating specifications and main dimensions of the RV Compressor

Operating specifications	
volumetric displacement, V_c	12.15 cm ³ /rev
operational speed, ω_{ro}	358 rad s ⁻¹
working fluid	R22
suction pressure, P_s (abs)	0.583 MPa
discharge pressure, P_d (abs)	2.03 MPa
Main dimensions	
rotor radius, R_{ro}	23.8 mm
cylinder radius, R_{cy}	27 mm
compressor axial length, L_c	23.8 mm
rotor bearing radius, $R_{b,ro}$	9.6 mm
total length, $L_{b,ro}$	58 mm
radial clearance, $\delta_{b,ro}$	13 μ m
cylinder bearing radius, $R_{b,cy}$	14.8 mm
total length, $L_{b,cy}$	34 mm
radial clearance, $\delta_{b,cy}$	13 μ m
Assembly radial clearance, δ_o	20 μ m
lubricant dynamic viscosity, μ	3.4 mPa s