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Theoretical Study of a Novel Multi Vane Rotary Compressor

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ABSTRACT

This paper presents results of a theoretical study on a new novel concept of a multi-vane rotary (MVR) compressor with the axis vertical. In this concept a sleeve (or cylinder) and a mechanical rotor are eccentrically mounted such that both touch each other circumferentially at one linear point of contact during a rotation. One vane (primary) has its rounded end embedded fairly loosely into the sleeve causing it to be rotated along with the vane and at the same time the vane reciprocates in a slot in the mechanical rotor. The other vanes (secondary) are free to slide out of their respective slots in the rotor, by centrifugal force and are similarly pushed in by the rotating sleeve. With this arrangement compression is guaranteed at all speeds. As soon as the rotor begins to rotate suction and compression chambers are respectively created by the primary vane, while the secondary vanes only start performing the compression function when enough centrifugal force has been built up. In addition, the relative rotation between the sleeve and each secondary vane reduces rubbing between the two. With this arrangement the study found out that the performance of the compressor is significantly improved compared to other similar existing refrigerant compressors of multi-vane rotary type. The reduction in friction is estimated to be about 45%. Analysis of instantaneous internal leakage was also conducted and was found that this new compressor concept possesses a relatively high volumetric efficiency. The work is currently continued involving designing, fabricating and performance testing of the compressor.

1. INTRODUCTION

In a family of MVR compressors, mechanical and volumetric efficiencies are the two most important parameters to be improved. Mechanical efficiency decreases with the increase of mechanical friction that occurs on respective rubbing surfaces in the machine. The frictional force can simply be calculated using a Coulomb law. The energy required to overcome this resistance is dependent on the coefficient of friction and the relative velocity of the rubbing surfaces. In a typical rotary vane compressor frictions occur at the tip, end and side surfaces of the vane, end surfaces of the rotor, journal bearing and at an eccentric rotor-sleeve contact line. As investigated experimentally by Kaiser and Kruse (1984), the mechanical efficiency measured for a multi-vane compressor was about 60% to 82% of it shaft power at the respective speeds between 2000 to 6000 rpm.

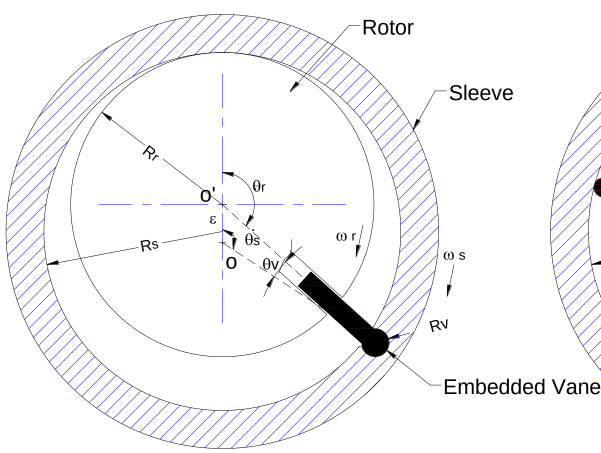
Volumetric efficiency of the compressor is the ratio of actual mass of gas discharged at exit condition to the theoretical mass that enters the compressor at the suction condition. Sarip (2005) and Kaiser and Kruse (1984) studied the performance of an MVR compressor experimentally and found that the compressor has volumetric efficiency ranging from 40% to 70%. This performance is generally still low compared to that of a single vane rotary (SVR) compressor which can achieve a performance of up to 90% volumetric efficiency. Nevertheless, for a same ratio of rotor diameter to inner diameter of sleeve, the MVR compressor virtually gives a bigger swept volume (cell volume at end of suction times number of vanes). Experimentally the actual mass can easily be measured directly using a flow meter or be calculated using heat balance of the gas and secondary fluid in a calorimeter. Theoretically, the volumetric efficiency value is attributed by the size of the clearance volume, the amount of re-expansion of carried over gas during suction process and the rate of net instantaneous internal leakage in a cell chamber. As shown in Figure 3, the clearance volume for an MVR compressor is located after the discharge port where the gas is trapped, but at a certain high pressure the trapped gas is forced to flow through to the suction chamber and mixes with the fresh gas that enters the chamber through the suction port. The high temperature carried over gas heats up the mixture causing the suction pressure to drop slightly. With this condition the specific density of the gas can increase roughly by about 15% to 20%, at speeds below 3000 rpm. The effect of this carried over trapped gas is a reduction in the volumetric efficiency which can be about 3% to 5%. The net instantaneous internal leakage is caused by the gas that leaks in from the leading cell and by the gas that leaks out to the trailing cell. The estimated further reduction in the volumetric efficiency due to internal leakages can be about 5% to 15%, at speeds below 3000 rpm.

The discussions above are about the effect of friction to a mechanical efficiency and about the effects of internal leakages and carried over trapped gas to a volumetric efficiency. These two efficiencies must be improved for MVR compressors to compete with that of scroll and/or rolling piston types. Many new designs of rotary multi vane compressor have emerged posing new challenges but none has yet able to show presence commercially, let alone to replace the presence of the traditional multi vane compressor models. The development of this new RSMV compressor, which has been studied analytically, is hoped to change the scenario. The study focused on the total friction and on instantaneous internal leakages. The effect of the expansion of carried over trapped gas on the volumetric efficiency will be studied later.

2. OVERVIEW OF RSMVR COMPRESSOR

In year 2001, Musa (2007) has developed a new concept of single vane compressor namely, rotating sleeve single vane rotary (RSSVR) compressor. As shown in Figure 1(a), one end of the vane is embedded into the sleeve and during a rotation the other part of the vane is pushed in and pulls out of a slot in the rotor. The primary function of the vane is to drive the sleeve to rotate along and together with the rotor all components perform the compression process. A patent was filed in the same year and granted in year 2007. There are also other similar concepts, some are modifications to this model, are being developed and studied by other researchers. Examples of such concepts are the synchronal vane by Zong Chang and Xin Wei (2004), the revolving vane by Teh and Oii (2009a). Figure 1(a) shows the typical diagram of RSSVR compressor. Evolved from this single primary vane of embedded end model, a multi-vane concept is introduced. Unlike the primary vane the function of these secondary vanes is to perform compression only. The invention is called a rotating sleeve multi vane rotary (RSMVR) compressor and the concept is as shown in Figure 1(b). The thermodynamic process in this RSMVR compressor concept is similar to that in a conventional MVR compressor.

One advantage of the multi vane concept is that the swept volume is virtually increased to about twice that of the RSSVR. For example, if the RSSVR compressor has radius of rotor 30 mm, inner radius of sleeve 34.5 mm and length 32.6 mm, then the swept volume is 32.46 cc. If four secondary vanes are added to become an RSMVR compressor, then the virtual swept volume is 66.63 cc. Thus, if a swept volume is specified as 32.46 cc adopting an RSMVR concept makes the compressor smaller, compact and lighter.



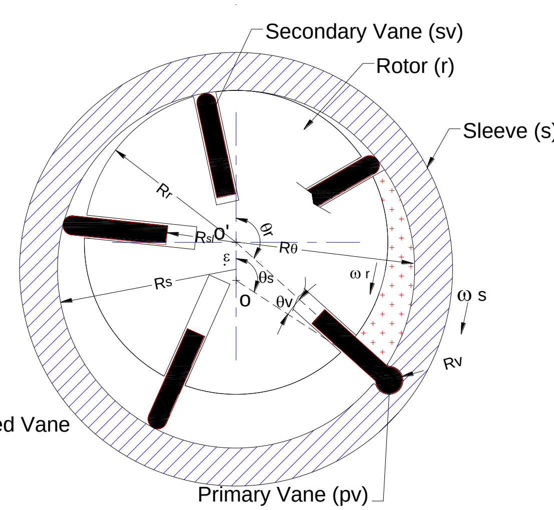


Figure 1(a): Sectional view of RSSVR compressor   Figure1 (b): Sectional view of RSMVR compressor

Furthermore, the eccentricity between the rotor and the rotating sleeve creates an interesting movement of each secondary vane in relation to the sleeve. When centers of rotor, any secondary vane and sleeve are all three in line, as shown in Figure 2, the rubbing action of the sliding vane with the surface of the sleeve is in the direction of rotation and occurs within a certain small angle. However, when the same three centers are in line for the second time within the same revolution, the rubbing action is in the opposite direction. The friction on the tip of each secondary vane is estimated to be small and the rubbing angle of the portion on the sleeve that each secondary vane rubs is about 15º. Whereas, in a conventional MVR compressor, the vane tip is rubbing against the cylinder wall throughout the rotation (or 360º). As the rotor rotates, the sleeve will push and pull the primary vane into and out of the slot in the rotor, respectively. The other four vanes are similarly pushed into the respective slots in the rotor but slide out only when a sufficient centrifugal force is developed. The primary vane is one of the most important components in this compressor in which the rotation of the sleeve is totally dependent on it. The design must be rigid to take all the loads which increase with the increase of the gas pressure, vanes dynamics forces and bearings friction torque. In this concept, the primary vane ensures that suction, compression and discharge of the gas occur at all speeds of rotation as shown in Figure 3 which also shows that at certain low speeds all secondary vanes do not participate in the compression process. In addition, at the same low speeds there is no leakage around the tip of the secondary vane and this will give a higher volumetric efficiency. In general, the RSMVR compressor is supposed to perform better compared to the existing MVR compressor, in term of mechanical and volumetric efficiencies.

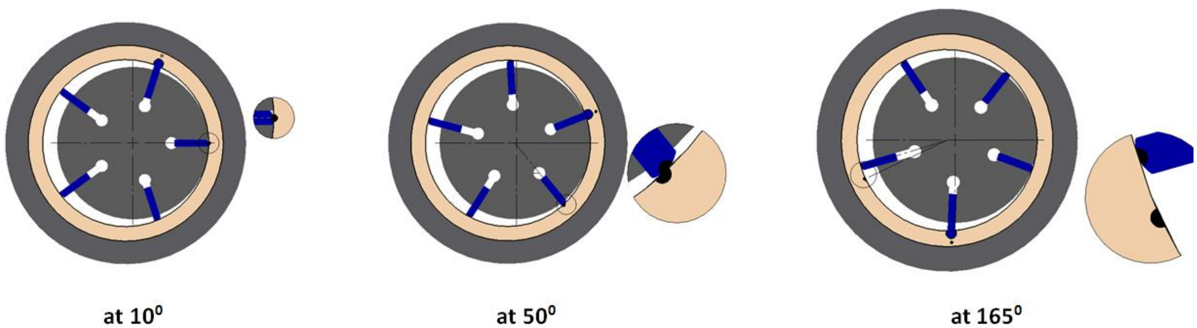


Figure 2: Rubbing surfaces of secondary vane and of sleeve

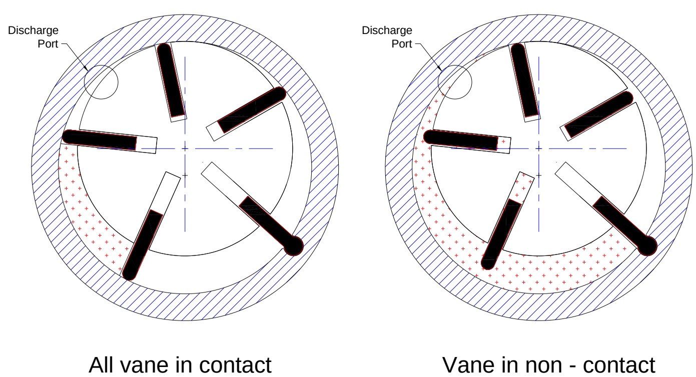


Figure 3: Comparison of two positions of each secondary vane tip – high speed and low speed

3. THEORETICAL DEVELOPMENT

Theoretical analysis of friction and instantaneous leakages has been studied by few researchers for MVR and RSSVR compressors respectively. For example, a mathematical model used to conduct a dynamic analysis on a radial vane rotating within a circular cylinder has been developed by Chang (1983) who presented the results on the frictional loses. A similar study was done by Teh and Oii (2009a,b) but for friction and internal leakage analysis of single vane compressor.

To start the friction analysis, cell volume distribution and kinematics as well as dynamics of components must first be established. All reaction forces need to be identified. Therefore for the RSMVR geometry, the ideal cell volume distribution via the leading vane of the compressor is as shown in Figure 1(b) and the distribution can be modeled mathematically by Equation (1). The volume effects of vanes and oil film are neglected.

$$V_c(\theta_r) = \frac{\Gamma R_s^3}{4}\begin{bmatrix} 2(\sigma - \Phi_1 + \Phi_2) \\ +\zeta^2(\sin(2\theta_r) - \sin(2\theta_r - 2\sigma) + (sin2\Phi_1 - sin2\Phi_2) - 2\Omega^2\,\sigma \end{bmatrix} \qquad (1)$$

where,

$$\Phi_1 = \sin^{-1}(\zeta\sin(\theta_r))\,,\ \Phi_2 = \sin^{-1}(\zeta\sin(\theta_r - \sigma))$$

$$\sigma = \frac{2\pi}{N_v}, \zeta = \frac{\epsilon}{R_s}, \Omega = \frac{R_r}{R_s}\,, \Gamma = \frac{L}{R_s}$$

In this analysis, due to eccentricity between the sleeve and the rotor and the structural connection of the two by a primary vane, the sleeve is selected to be assumed to rotate at a steady state with a constant angular velocity while the velocity of the rotor fluctuates. Inversely, the assumption is also true. With the sleeve rotating at a constant velocity the rotor angular fluctuating velocity is described mathematically by Equation (2). All vanes are rotating and simultaneously sliding radially to and from the centre of the rotor. The kinematics of the vanes in rectilinear motions can be described by Equation (3). The rectilinear motion of primary and secondary vanes is assumed comparatively same. Furthermore the primary vane swivels about the embedded rounded end allowing the system to rotate smoothly. However, the swivel velocity can be neglected as the effect is very small compared to that of the rotor angular velocity. The friction between the tip of each secondary vane and the surface of the sleeve depends on their relative angular velocity which is given in Equation (4).

$$\omega_r = \frac{d\theta_r}{dt} = \frac{d\theta_r}{d\theta_s} \times \frac{d\theta_s}{dt} \qquad (2)$$

$$v_v = \frac{dR_\theta}{dt} = \frac{d\theta_r}{dt} \times \frac{dR_\theta}{d\theta_r} \qquad (3)$$

$$v_{rel} = R_v(\omega_s - \omega_r) \qquad (4)$$

3.1 Friction Analysis

The major friction on the RSMVR compressor exists on the pairing contact surfaces of vane-sleeve, vane-rotor and rotor-sleeve respectively, as shown in figure 4(a),(b) and (c). The friction that occurs on primary and secondary vanes actually can be modelled at different modes as described by Chang (1983). In this analysis the contact forces induced by primary vane is at both sides of the vane and that induced by the secondary vanes are at the sides and tip of each vane.

The dynamic components that can also be seen in Figure 4 are the Coriolis, inertia and centrifugal forces which act on the centre of gravity of each vane. The pressure forces are shown to act on the sides of vane and on the contacting surfaces of rotor and sleeve. The friction torque on journal bearing and end surface of rotor and sleeve are not shown here. With reference to Figure 4, all contact forces will generate friction power and can be calculated using the following relations:

$$\begin{aligned} P_f = {} & \left[\mu_f\left(\left|R_{pv,N}\right|\omega_v + (\left|R_{pv,1} + R_{pv,2}\right|)v_v\right)\right]_{priary\ vane} \\ & + \left[\left[\mu_f\left(\left|R_{sv,N}\right|v_{rel} + (\left|R_{sv,1} + R_{sv,2}\right|)v_v\right)\right](N_v - 1)\right]_{secondary\ vane} \\ & + \left[T_{e,r}\omega_r + T_{e,s}\omega_s\right]_{end\ surface} \\ & + \left[\mathrm{T}_{b,r}\omega_r R_{b,r} + \mathrm{T}_{b,s}\omega_s R_{b,s}\right]_{journal\ bearing} \end{aligned} \qquad (5)$$

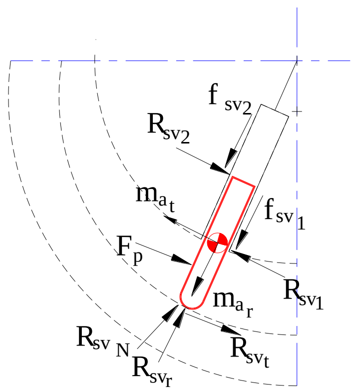


Figure 4(a): Acting force on secondary vane

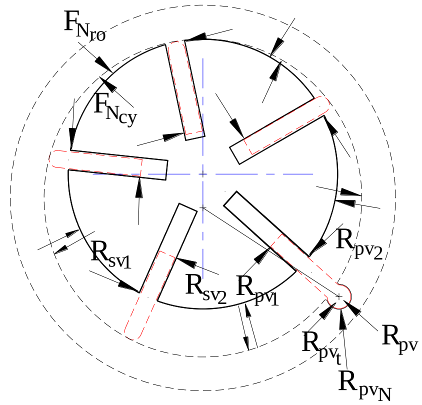


Figure 4(b): Acting force on rotor-sleeve

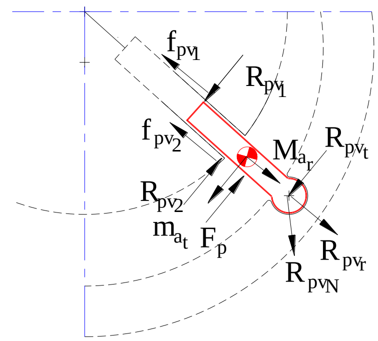


Figure4(c): Acting force on primary vane

The analysis begins by determining all contact and friction forces on secondary vane as shown in Figure 4(a). The method of analysis has been discussed briefly by Chang (1983). Then all contact and friction forces from secondary vane will transfer to the general analysis of rotor and sleeve as shown in Figure 4(b) to determine the resultant force that acts on sleeve and rotor. Nevertheless forces resulting from vane friction are initially to be neglected as it is assumed small compared to pressure and contact forces. The resultant force acted on the sleeve and rotor will be used for journal bearing analysis which is to determine the character of the bearing as well as torque friction which can be modelled using a simple method of combination of short and long journal bearing as described by Hirani *et al*. (1997). The friction force relation as expressed in Equation (6) where the F_{bx} and F_{by} are a total force acting in a Cartesian coordinate system in x-y axis on the sleeve and rotor respectively.

$$\mathrm{T}_b = \frac{\mu \omega_b R_b^2 L_b \pi}{c_b \sqrt{1-{\varepsilon_b}^2}} \left(\frac{2+\varepsilon_b}{1+\varepsilon_b} \right) + \frac{c_b \varepsilon_b}{2 r_b} \sqrt{F_{b,x}^2 + F_{b,y}^2} \sin \Phi \qquad (6)$$

Both rotor and sleeve are rotating in the same direction. Hence the friction between the two at the eccentric contact line is not significant. However, the frictions between the respective end surfaces of the rotor and sleeve with the two opposite end plates are significant. These frictions can be modelled using a simple Petroff's Law. The surface clearance between rotor/sleeve and the upper end plate is assumed equal with that between rotor/sleeve and the bottom end plate. In reality the axial surface clearance with the bottom end plate is smaller due to gravity exerted by the masses of the rotor, vanes and sleeve, respectively. The relation of torque friction from end surface can be expressed as follows:

$$\mathrm{T}_{r/s} = \frac{\mu}{c_e} \frac{\pi}{2} \omega_{r/s} \left[R_o^4 - R_{in}^4 \right] \qquad (7)$$

Lastly is the analysis on primary vane. The rotating mechanism of RSMVR compressor is totally dependent on the primary vane. It will take all loads from the contact force, force and torque frictions and pressure force to drive the sleeve as well as to create compression. Obviously the primary vane must be stronger than the secondary vane. Figure 4(c) shows the four contact forces exist on the vane. To determine this four unknown forces, four equations need to be derived using moment and force balance and then analyze them simultaneously. The analysis begins by taking all moments or torques about the centre of the sleeve and of the rotor as described in Equation (8) and Equation (9). In this analysis moment resulting from secondary vane tip friction force is neglected as it is assumed to be a very small. Hence, only moments due to respective contact forces formed by side friction on secondary vane, torque friction of journal bearing and on end surfaces of sleeve and rotor, are included in the dynamic analysis.

$$\sum M_{o_s} : R_{pv,r}\varepsilon \sin(\theta_r) + R_{pv,t}R_s \sin\left(\theta_s + \frac{\pi}{2}\right) = I_s\alpha_s + \mathrm{T}_{b,s} + \mathrm{T}_{e,s} \tag{8}$$

$$\sum M_{o_r} : R_{pv,1}R_{sl} - R_{pv,2}R_r = I_r\alpha_r - \sum f_{f,v,1}R_{sl} + \sum f_{f,v,2}R_r + \mathrm{T}_{b,r} + \mathrm{T}_{e,r} \tag{9}$$

The analysis continues by taking all forces that exist on the primary vane in r-t direction as describe in Equation (10) and Equation (11) which are derived using the second law of Newton for a curvilinear motion. Further, the four unknowns can be solved by simultaneous linear equation, combining all Equation (8) to Equation (11) in matrix form or in other mathematical forms. An iteration procedure can be conducted to calculate all the four unknown contact forces that exist during one complete rotation.

$$\text{r direction} -: \mu_f R_{pv,1} - \mu_f R_{pv,2} - R_{pv,r} = m(a_v - R_\theta {\omega_r}^2) \tag{10}$$

$$\text{t direction} -: R_{pv,1} + R_{pv,2} - R_{pv,t} = m(R_\theta\alpha_r + 2v_v\omega_r) + \mathrm{F_p} \tag{11}$$

Figure 5 shows the variation of the contact forces on the primary vane for one complete working cycle. The side contact forces peak at about 280N which is about 40 % higher than secondary vane side forces in the vicinity of compression process while simultaneously the force at the tip of the vane is below 30N. As expected the sides of primary vane have a dominating contact force due to its primary function in taking all loads to rotate the sleeve and at the same time to perform compression.

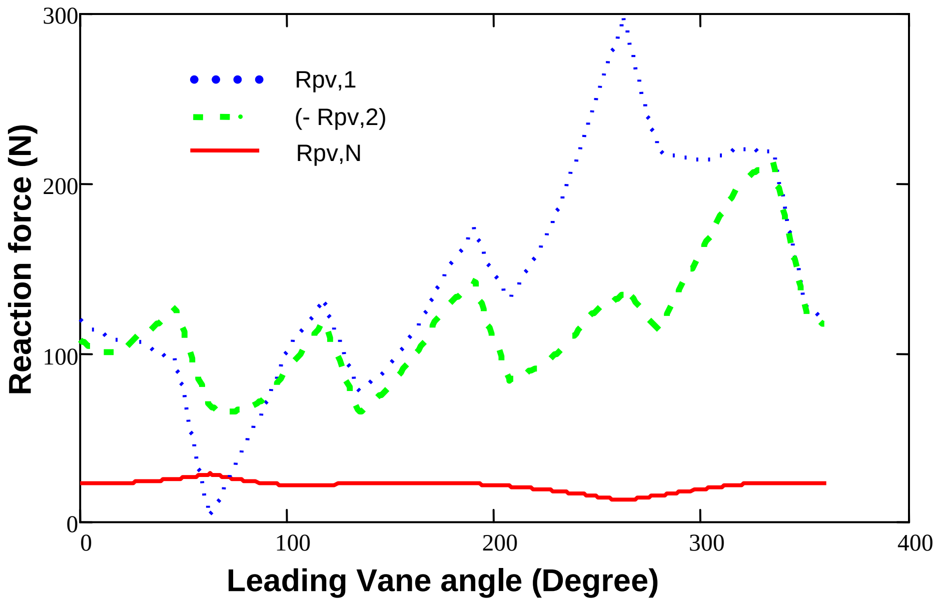


Figure 5: Reaction force at side and hinge of primary vane

3.2 Cell Leakage Analysis

RSMVR compressor has two types of leakage flows from a high pressure cell to a low pressure cell. The leakage can be both axial and radial. The axial leakage type is where the flow passes through the axial clearance between the flat surfaces of two components. For the RSMVR compressor the axial clearances are at the two end surfaces of each vane, sleeve and rotor respectively. While the radial leakage type the flow passes through the radial surfaces of two components and can be found at tip of each secondary vane and at the eccentric contact line of rotor and sleeve.

Yanagasiwa and Shimizu (1985) found the volumetric efficiency decrease is negligible when the end face clearance is below 10 μm. In line with this finding and due to gravity, the sleeve, rotor and side vane bottom end face clearances are all assumed below 10 μm and the effect of these clearances is neglected in the analysis. The compressor is designed with the axis vertical. Radial leakage of trapped gas in the discharged cell through the eccentric contact line to the suction cell is assumed steady and uniform along the contact line and the pressure difference between these two cells is taken as half of the discharged pressure. In fact in real situation the clearance along the eccentric rotor/sleeve contact line fluctuates due to the dynamics radial clearance of the journal bearing characteristic. This eccentric contact line leakage is added to the leakage in the opposite direction which is from a cell of maximum pressure through cells of intermediate pressures and finally to a cell of minimum pressure. The entire leakage can be modeled as shown in Figure 6 and the mass flow rate of this internal leakage is calculated as follows:

$$\dot{m}_L = [\dot{m}_r]_{eccentric} + [\dot{m}_a]_{primary\ vane} + \left[\sum(\dot{m}_a + \dot{m}_r)\right]_{secondary\ vane} \qquad (12)$$

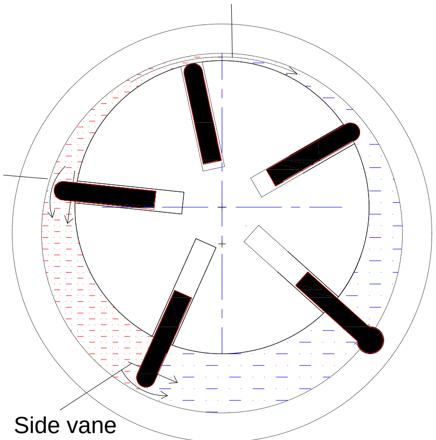


Figure 6 : Internal leakage path

The method of analysis for leakage is using an incompressible and viscous flow model. The model has been used by Ishii *et al*. (1998) to simulate the performance (volumetric efficiency) of a scroll compressor. The formula was originally used by Darcy, a French engineer, for calculating head loss in pipe. The general equation of leakage analysis according to Darcy is as follows:

$$\frac{\Delta P}{\rho g} = f_L \frac{L_L}{4m} \frac{V_L^2}{2g} \qquad (13)$$

The friction factor, f_L is assumed to be for turbulent flow for entire leakage path, which can be expressed as follows:

$$f_L = 0.35 R_e^{-0.25} \qquad (14)$$

Leakage path for eccentric rotor-sleeve clearance and tip clearance of secondary vane can be calculated using the following expression:

$$L_L = \frac{2\pi c_L R_s}{(R_s - R_r)\left(\sqrt{1 - (R_s - R_r)^2}\right)} \qquad (15)$$

4. RESULT AND COMPARISON

The comparison of both MVR and RSMVR compressor is based on same design operating parameters and dimensions of main components as shown in Table 1.

Table 1: Design parameters and main dimensions

Design Parameters			
Volumetric displacement	50.0 cc	Suction Temperature	278 K
Cooling capacity	4.1 kW	Discharge temperature	353 K
Operational speed, ω_c	3000 rpm	Suction pressure	0.3 Mpa
Working fluid	R134a	Discharge Pressure	1.5 Mpa
Main Dimensions			
Rotor radius, R_r	31.0 mm	Axial Length, L_c	36.0 mm
Inner sleeve radius, R_s	34.5 mm	Vane Height, H_v	14.0 mm
Outer sleeve radius, R_{st}	40.5 mm	Number of vane, N_v	5 no.

Figure 7 shows typical graph for a total friction power which is generated at the rounded-end of the primary vane, tips of secondary vane and at the sides of all vanes. The value of coefficient of friction is 0.1. The friction shows a trend of increasing power with the angle of rotation, reaches maximum at certain leading vane angle, after which the power decreases. For MVR compressor the average friction power is about 375 Watt while for RSMVR compressor the average friction power is about 62 Watt. Comparing the two graphs it is seen that the friction power peaks to a value of about 585 Watt at about 270^{o} for the RMV compressor and to a value of about 140 Watt at about 185^{o} for the RSMVR compressor. The frictional wear can therefore be very high for an MVR compressor and over a long period of operation the rate of wear can be critical at an angle where the friction power peaks. For the RSMVR compressor the rate of frictional wear on tip of secondary vane is reckoned to be insignificant.

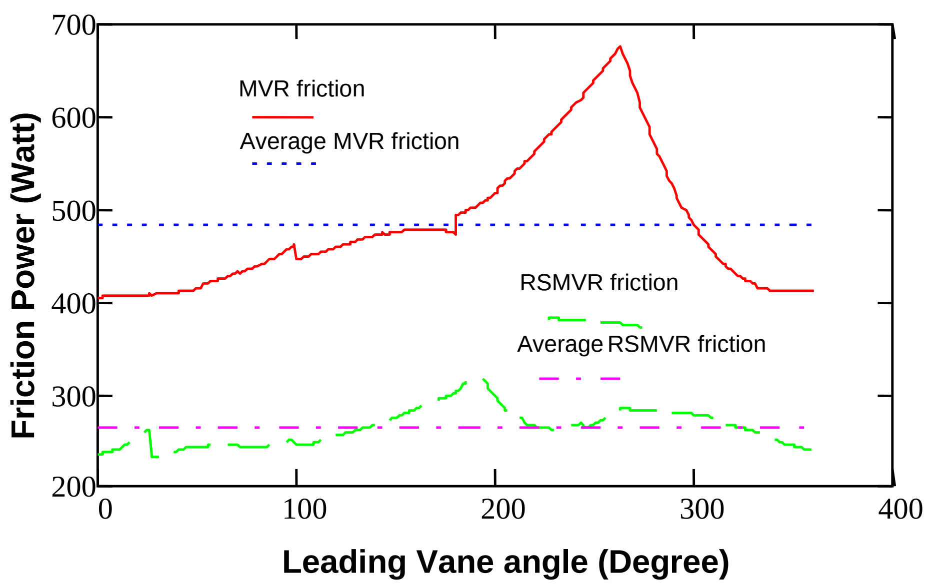


Figure 7: Variation of total friction losses at N= 3000 rpm

Figure 8 shows the variation of mass inside the cell of RMV and RSMV compressors respectively, during compression and discharge processes. The clearance of each leakage path is assumed constant during one complete rotation. However, during each rotation, the clearance between rotor and sleeve at the eccentric contact line, as set during assembly, may change due to the dynamics of the journal bearing. In the analysis the clearance is set at 20 μm. The clearance around the swinging tip of the primary vane is 5 μm and those between both sides of all vanes and walls of respective slots in the rotor, are each 10 μm. The figure shows that the mass inside the cell increases during compression process then rapidly decreases during discharge process. Taking one particular cell as reference, the leakage on the leading cell is more severe compared to that on the trailing cell due to different in pressure differences at adjacent to the investigated cell. While at the eccentric contact line the leakage shows to be more severe because of the maximum pressure difference between discharge and suction chambers (or cells). The gas will expand to the low pressure chamber as shown in Figure 6 (from red to blue chambers) until the trailing vane passes the eccentric line, which is at about 72^{o}. The two graphs in Figure 8 show that the cell masses for MVR and RSMVR compressors are about the same and so are the respective variations with the angle of rotation. The calculated volumetric efficiencies of both compressors are about 91.5%

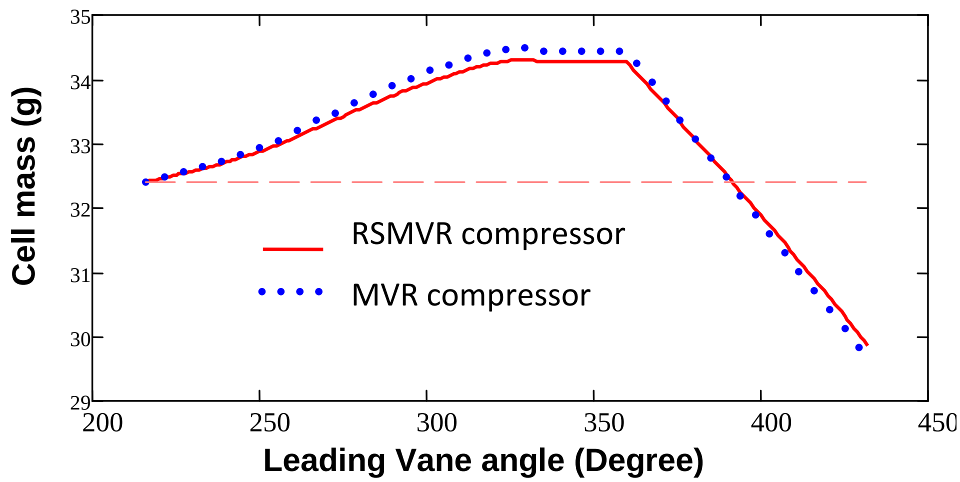


Figure 8: Variation of total mass in a cell during compression and discharge at pressure ratio 15

5. CONCLUSION

Friction and internal leakage on an MVR and on an RSMVR compressors respectively, have been studied. The analysis shows that the average power to overcome the total friction created by the vanes in relation to the sleeve and rotor is about 45% higher in MVR compressor as compared to that in RSMVR compressor. The power can be up to 585 Watt at a critical lead vane angle of 270^{o}, for the MVR compressor. This means that between the two compressors, RSMVR has a higher mechanical efficiency. The result also shows that the concept of primary vane that drives the sleeve to rotate along does not give any impact on the instantaneous internal leakage. The analysis finds the volumetric efficiencies are about the same for both MVR and RSMVR compressors. Nevertheless, the small relative velocity between tip of each secondary vane and surface of sleeve will result in less rate of wear and tear which in the case of MVR compressor, the intense rubbing between the vanes and surface of cylinder enhances leakage. As such, the RSMVR compressor will have an extended life span as compared to that of the MVR compressor.

NOMENCLATURE

F_p	pressure different	(Pa)	**Subscripts**	
R_e	Reynolds number	(-)	a	axial
$\dot{m}$	mass flow rate	(kg/s)	b	bearing
T	torque	(Nm)	e	end surface

I	mass moment inertia	(kg m^2)	In	inner
L	lenght	(m)	L	leakage
M	moment	(Nm)	o	outer
P	power	(Watt)	pv	primary vane
R	reaction force	(N)	r	rotor,radial
V	volume	(m^3)	rel	relative
c	clearance	(kg/s)	s	sleeve
f	force	(N)	sv	secondary vane
g	gravity	(m/s^2)	f	friction
m	mean diameter	(m)	sl	slot
p	pressure	(Pa)		
v	velocity	(m/s)		
η_m	mechanical efficiency	(-)		
η_v	volumetric efficiency	(-)		
μ_f	coefficient of friction	(-)		
α	angular acceleration	(rad/s^2)		
ρ	density	(kg/m^3)		
ω	angular velocity	(rad/s)		

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