Investigation of the Centrifugal Force Effect to a Revolving Vane (RV) Machine

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Investigation of the Centrifugal Force Effect to a Revolving Vane (RV) Expander

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ABSTRACT

The Revolving Vane (RV) machine is unique because the cylinder rotates together with the rotor at their respective axis. This causes various unique behaviors that are not previously observed in the conventional rotary machines. In this paper, the effects of the centrifugal force to the behavior of the machine are investigated. It is found that the centrifugal force affects the average output torque through the shaft bearing load and friction loss. Interestingly, if the center of mass is set properly, the centrifugal force can help to offset the pressure force acting at the shaft. This helps to reduce the bearing load and increases the mechanical efficiency of the machine. The finding can be used to optimize the RV machine design.

1. INTRODUCTION

Due to the recent demand of more energy efficient refrigeration systems, expanders have been intensively researched in the past few years in place of the conventional expansion valve. Simply speaking, an expander is a compressor that operates in the reversed cycle. It takes in high pressure fluid and expands it to produce power. The generated power can then be used to help to run the compressor. At the same time, the slow expansion process reduces the throttling loss by making the process closer to the isentropic line, increasing the cooling capacity.

One of the first proposed applications of expanders is for transcritical carbon dioxide refrigeration systems where due to the large pressure difference used, the throttling loss is significantly large. The early studies indicate that an expander with a 60% isentropic efficiency can increase the coefficient of performance (COP) of the CO_2 system by around 20% (Robinson and Groll, 1998). Since then, various types of expanders have been developed and studied based on the existing compressor mechanisms. These include, among others, reciprocating (Baek et al., 2005a, b), scroll (Kim et al., 2008), screw (Kovacevic et al., 2006), rolling piston (Matsui et al., 2009) and rotary vane (Fukuta et al., 2009). More recently, a new type of rotary mechanism, called the Revolving Vane (RV) mechanism, where the cylinder rotates together with the rotor was introduced (Teh and Ooi, 2006). This unique feature allows the mechanism to have less relative velocities at the rubbing surfaces, resulting in smaller friction losses. The mechanism has then been improved by attaching the vane rigidly to the rotor (Teh and Ooi, 2008b) and also used for expander applications (Subiantoro and Ooi, 2009). An illustration of the working process of the RV expander is shown in Figure 1.

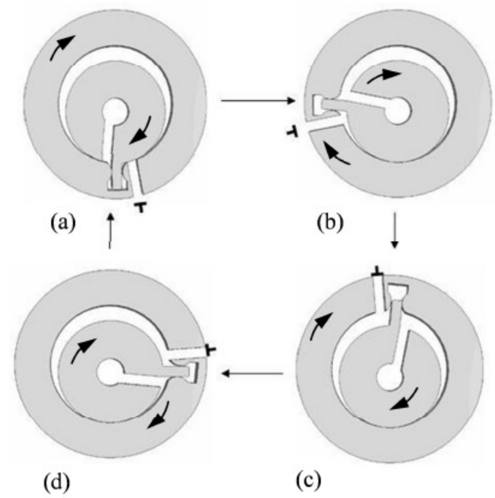

Figure 1: Working process of the RV expander

Various aspects of the mechanism have been studied, these include the friction losses (Subiantoro and Ooi, 2011; Teh and Ooi, 2008b), internal leakages (Teh and Ooi, 2008a) and vane arrangement (Subiantoro and Ooi, 2010). In this paper, the effects of the centrifugal forces of the components are analyzed. The analysis will be focused on the effect to the expander torque production.

2. THEORETICAL MODEL

The study is carried out mathematically. For simplification, the following assumptions are made:

- Adiabatic process is assumed for all processes.
- Perfectly sealed model is assumed.
- The driving shaft operates at a constant angular velocity.
- The frictional losses only occur at the shaft bearings, vane side and endfaces, since other frictional losses are negligible.

The mathematical models include the thermodynamics, kinematics and dynamics aspects of the expander. They are solved with the Runge-Kutta method using a computer code written in the Fortran programming language. The thermodynamics model, which simulates the thermodynamics of the working fluids in the two working chambers, is solved first based on equation (1), which is the energy balance equation. The results are then used to simulate the dynamics aspects based on equations (2)-(7). They are derived from the free body diagrams of the rotor and the cylinder. Detailed presentations of the derivations can be found elsewhere (Subiantoro and Ooi, 2009). They include the calculations of the expander output torque, T_{gen}, the vane contact force, $F_{v,n}$, and the shaft bearings loads, F_b. Centrifugal forces occur at both the rotor and the cylinder and affect the performance of the expander through the shaft bearing loads.

$$m_{cv}\frac{du_{cv}}{dt}+u_{cv}\frac{dm_{cv}}{dt}=\frac{dQ_{cv}}{dt}-\frac{dW_{cv}}{dt}+\sum_i\left(h_i+\frac{v_i^2}{2}\right)\frac{dm_i}{dt}-\sum_o\left(h_o+\frac{v_o^2}{2}\right)\frac{dm_o}{dt} \tag{1}$$

$$T_{gen}=F_{v,p}\left(r_r+\frac{l_{ve}}{2}\right)-F_{v,n}\left(r_r+l_{ve}+\frac{dl_{ve}/dt}{\left|dl_{ve}/dt\right|}\eta_v\frac{w_v}{2}\right)+\sum T_r \tag{2}$$

$$F_{v,n}=\frac{I_c\alpha_c-\sum T_c}{r_c\cos\gamma-\frac{F_{v,n}}{\left|F_{v,n}\right|}\frac{dl_{ve}/dt}{\left|dl_{ve}/dt\right|}\eta_v\left(r_c\sin\gamma-\frac{F_{v,n}}{\left|F_{v,n}\right|}\frac{w_v}{2}\right)} \tag{3}$$

$$F_{b,x,c} = -F_{v,n}\cos\varphi_r + F_{v,f}\sin\varphi_r - F_{p,c}\sin\frac{\varphi_c}{2} - F_{ct,c}\sin(\varphi_c + \theta_{ct,c}) + F_{loss,x,c} \tag{4}$$

$$F_{b,y,c} = F_{v,n}\sin\varphi_r + F_{v,f}\cos\varphi_r - F_{p,c}\cos\frac{\varphi_c}{2} - F_{ct,c}\cos(\varphi_c + \theta_{ct,c}) + F_{loss,y,c} \tag{5}$$

$$F_{b,x,r} = -F_{v,p}\cos\varphi_r + F_{v,n}\cos\varphi_r - F_{v,f}\sin\varphi_r + F_{p,r}\sin\frac{\varphi_r}{2} - F_{ct,r}\sin(\varphi_r + \theta_{ct,r}) + F_{loss,x,r} \tag{6}$$

$$F_{b,y,r} = F_{v,p}\sin\varphi_r - F_{v,n}\sin\varphi_r - F_{v,f}\cos\varphi_r + F_{p,r}\cos\frac{\varphi_r}{2} - F_{ct,r}\cos(\varphi_r + \theta_{ct,r}) + F_{loss,y,r} \tag{7}$$

where:

$$F_{ct,c} = m_c r_{ct,c} \omega_c^{\ 2}$$

$$F_{ct,r} = m_r r_{ct,r} \omega_r^{\ 2}$$

$$F_{p,c} = (p_{scv} - p_{dcv}) 2 r_c l_{\exp} \sin(\varphi_c / 2)$$

$$F_{p,r} = (p_{scv} - p_{dcv}) 2 r_r l_{\exp} \sin(\varphi_r / 2)$$

$$F_{v,p} = (p_{scv} - p_{dcv}) l_{ve} l_{\exp}$$

The definitions of the angle and the radius of the center of mass, θ_{ct} and r_{ct}, are shown in Figure 2.

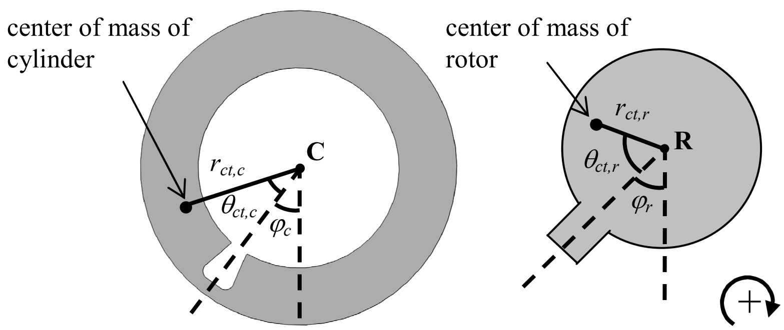

Figure 2: Schematic diagrams of the cylinder and the rotor

The main dimensions and parameters of the simulated expander model are listed in Table 1.

Table 1: Main dimensions and parameters of the expander model

Radius of rotor body	29 mm
Inner radius of cylinder body	35 mm
Length of expander	25 mm
Diameter of rotor shaft bearing	25 mm
Length of rotor upper shaft bearing	40 mm
Length of rotor lower shaft bearing	20 mm
Radial clearance of rotor shaft bearings	15 μm

Diameter of cylinder shaft bearing	49.2 mm
Length of cylinder shaft bearings	10 mm
Radial clearance of cylinder shaft bearings	30 μm
Rotating inertia of rotor	0.21 g.m^2
Rotating inertia of cylinder	2.81 g.m^2
Friction coefficient of vane	0.15
Discharge coefficient of ports	0.7
Lubricant viscosity	0.041 Pa.s
Diameter of suction ports (3 numbers)	6.4 mm
Diameter of discharge ports (3 numbers)	4.6 mm
Cylinder endface total gap width	100 μm
Mass of rotor	0.737 kg
Mass of cylinder	1.341 kg
Distance of center of rotor mass from rotor center	29 mm
Distance of center of cylinder mass from cylinder center	35 mm

3. RESULTS AND DISCUSSIONS

To assist in the observation of the effects of the centrifugal forces to the expander performance, a non-dimensional parameter called the Torque Difference, abbreviated as TD, is introduced. The definition is as shown in equation (8). T_{sim} is the output torque obtained from the simulation of the expander with the consideration of the centrifugal force(s). T_0 is the benchmark torque, which is equal to 0.782 Nm and is the output torque of the expander without any centrifugal force.

$$TD = \frac{T_{sim} - T_0}{T_0} \tag{8}$$

The effect of the direction of the centrifugal force is shown in Figure 3 through the relationship between TD and the angle of the centers of both the cylinder and the rotor masses. It can be seen that the centrifugal forces of the components can decrease the expander performance significantly. This is because the centrifugal forces add more loads to the shaft bearings, resulting in more losses at the bearings. The worst configurations are when the angles of the cylinder and the rotor centers of mass are at 0° and 180°, with TD values of around -28% and -8%, respectively. During the design stage, it is important to ensure that the centers of mass do not lie in the vicinities of these values. The centrifugal force of the cylinder is more significant in affecting the expander performance because the cylinder has more mass and is larger in size as compared to the rotor.

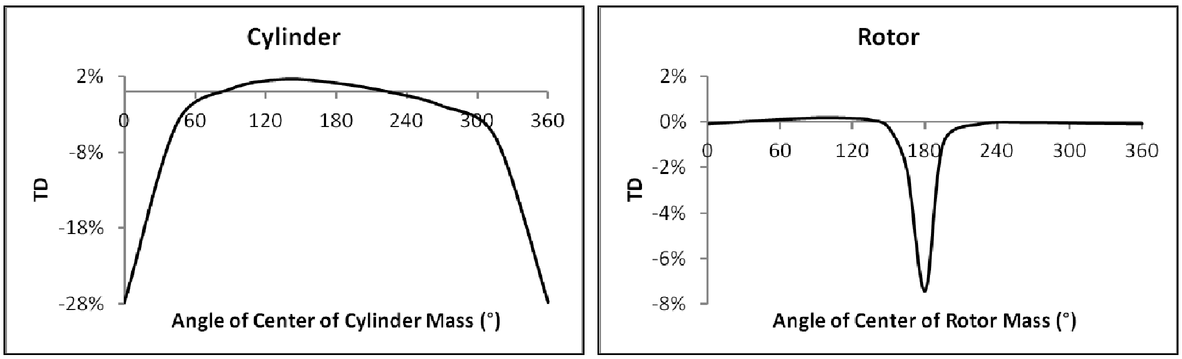

Figure 3: Relationship between TD and the angle of center of mass

Interestingly, Figure 3 also indicates that the centrifugal forces, especially those at the cylinder, can improve the expander performance by 2%. The best configuration is when the angle of the cylinder is around 135°. The improvement caused by the rotor centrifugal force is insignificant, less than 0.2%. To understand the reason behind this behavior, the comparison of the profiles of the output torque when the expander experiences no centrifugal force and when it experiences a centrifugal force at the cylinder with the angle of center of cylinder mass equals to 135° is shown in Figure 4.

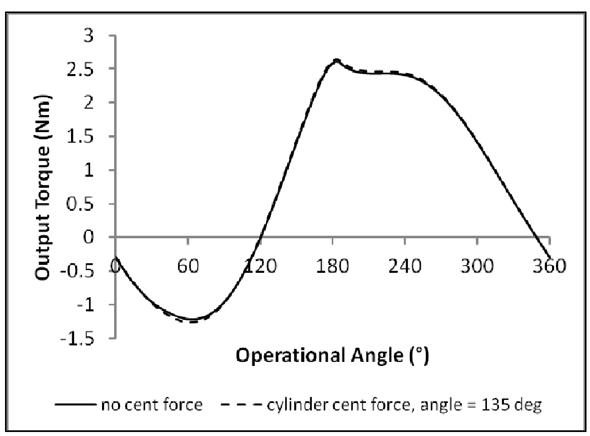

Figure 4: Comparison of output torque profiles of the expander with no centrifugal force and with the cylinder centrifugal force at an angle of cylinder mass of 135°

Figure 4 shows that when there is a centrifugal force at the cylinder where the cylinder mass is located at an angle of 135°, the amplitude of the expander output torque profile becomes larger than that of the expander with no centrifugal force. The negative peak is more negative because the centrifugal force adds more load the shaft bearing, resulting in more bearing friction loss. The positive peak is higher is higher in value because the centrifugal force helps to reduce the pressure load, resulting in less friction loss in the vicinity. This is illustrated in Figure 5. Because the pressure difference across the vane is at its highest here, this improvement is more significant than the decline of the negative peak, resulting in the overall expander performance.

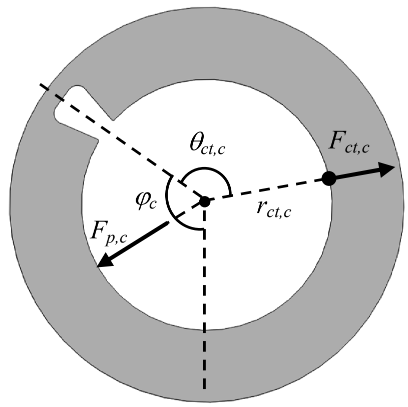

Figure 5: Interaction between centrifugal force and pressure load

Figure 5 shows the situation when the cylinder angle, φ_c, is around 135° and the angle of the center of the cylinder mass is designed as 135°. The pressure load at the cylinder is represented by $F_{p,c}$ and is acting in the direction towards the lower left hand corner of the diagram. On the other hand, the centrifugal force, $F_{ct,c}$, is acting in the opposite direction towards the upper right hand corner. Hence, the resultant force acting on the cylinder shaft is smaller than $F_{p,c}$. This causes the bearing load and its friction loss to decrease.

The effect of the magnitude of the centrifugal force of the cylinder to expander performance is shown in Figure 6. The magnitude is varied by varying the distance between the center of the cylinder mass and the cylinder center from 25 mm to 50 mm. It can be seen that there is an optimum distance, around 35 mm. However, the change in TD is small, less than 0.2%.

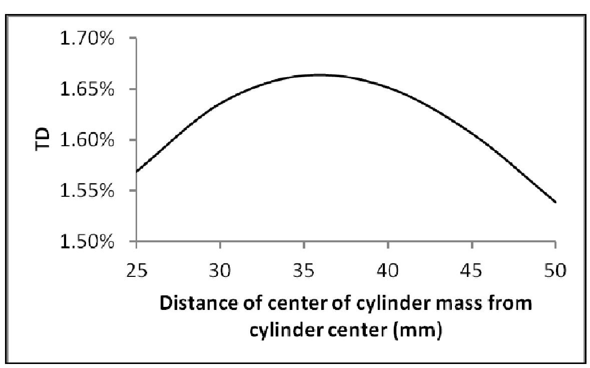

Figure 6: Relationship between TD and the distance of center of cylinder mass from cylinder center

4. CONCLUSION

In this paper, the effects of centrifugal force to the Revolving Vane expander performance have been studied. The performance indicator is the average output torque of the expander. From the study, the following findings are observed:

1. The centrifugal force of the cylinder affects the performance more significantly than the rotor because the cylinder has more mass and is larger in size, resulting in a larger centrifugal force.
2. The output torque of the expander can drop by as much as 28%. This happens if the center of mass of the cylinder is located near the line connecting the cylinder center and the vane slot (angle of center of center of cylinder mass of 0°).
3. From our study, the centrifugal force of the cylinder can improve the expander performance by about 2% if the center of the cylinder mass is located an angle of around 135°. The reason behind the improvement is because the centrifugal force reduces the bearing load when the pressure load is highest.
4. By varying the distance between the cylinder mass and the cylinder center, which results in the change of the centrifugal force magnitude, it can be observed that the optimum distance is at around 35 mm. However, the overall variation in the output torque is less than 0.2%.

NOMENCLATURE

F	force	(N)
I	rotating inertia	($kg.m^2$)
l	length	(m)
m	mass	(kg)
p	pressure	(Pa)
P	power	(W)
Q	heat energy	(J)
r	radius	(m)
t	time	(s)
T	torque	(Nm)
u	internal energy	(J/kg.K)
v	velocity	(m/s)
w	width	(m)
W	work	(J)
α	angular acceleration	(rad/s^2)
γ	angle between rotor and cylinder radii at vane contact	(rad)
φ	angle	(rad)
η	friction coefficient	(-)
θ	angle	(rad)
ω	angular speed	(rad/s)

Subscripts

b	bearing
c	cylinder
ct	centrifugal
cv	control volume
dcv	disc. chamber
exp	expander
f	friction
gen	generated
i	inlet
n	normal
o	outlet
p	pressure
r	rotor
scv	suct. chamber
v	vane
ve	exposed vane
x	x-axis direction
y	y-axis direction

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