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# Analysis of the Revolving Vane (RV-0) expander, part 2: Verifications of theoretical models

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ABSTRACT

Various aspects of the Revolving Vane (RV-0) expander have been theoretically modelled. They include the geometry, kinematics, dynamics, thermodynamics, flow through ports and internal leakage aspects. The models have also been compared to the experimental data. It was found that the predicted output torque matches with the experimental data with an average deviation of 20% while the flow rate predictions have an average deviation of 25%. The temperature of the journal bearing lubricant and the leakage flow rate through the housing clearances are thought to be the main sources of discrepancies. The study also found that the main contributor of the internal leakage is the flow through the radial clearance.



# Analyse du détendeur rotatif à palettes (RV-0), Partie 2 : vérifications des modèles théoriques

*Mots clés :* Simulation ; Modélisation ; Compresseur rotatif ; Détente ; Efficacité ; Énergie

## 1. Introduction

In recent years, studies about expanders have been picking up pace due to the urgent need to conserve energy, including in refrigeration systems. This is especially true in $CO_2$ refrigeration systems where the large throttling loss limits its use. Recovering the expansion work with expanders has been shown to be a way to overcome this problem (Lorentzen, 1994; Robinson and Groll, 1998).

Various types of expander have been studied (Fukuta et al., 2009; Guan et al., 2006; Kim et al., 2008; Kovacevic et al., 2006; Yang et al., 2009; Zhang et al., 2007), including the Revolving Vane (RV) type (Subiantoro and Ooi, 2009). The RV mechanism is different from the conventional rotary mechanisms

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**Nomenclature**

| | |
|---|---|
| $C_d$ | coefficient of discharge |
| $C_{lip\_seal}$ | tangential friction of the lip seal per unit circumference [N m$^{-1}$] |
| $d$ | diameter [m] |
| $E$ | total energy [J] |
| $e$ | eccentricity [m] |
| $F$ | force [N] |
| $g$ | gravitational acceleration [m s$^{-2}$] |
| $h$ | specific enthalpy [J kg$^{-1}$] |
| $I$ | inertia [kg m$^2$] |
| $L$ | total length [m] |
| $l$ | length [m] |
| $M$ | Mach number [–] |
| $m$ | mass [kg] |
| $P$ | power [W] |
| $p$ | pressure [Pa] |
| $Q$ | heat transfer energy [J] |
| $q$ | volume flow rate [m$^3$ s$^{-1}$] |
| $r$ | radius [m] |
| $R$ | gas constant [J kg$^{-1}$ K$^{-1}$] |
| $T$ | temperature [K]; torque [N m] |
| $t$ | time [s] |
| $u$ | specific internal energy [J kg$^{-1}$] |
| $V$ | volume [m$^3$] |
| $v$ | velocity [m s$^{-1}$] |
| $W$ | work done [J] |
| $w$ | width [m] |
| $z$ | height [m] |

*Greek letters*

| | |
|---|---|
| $\alpha$ | angular acceleration [rad s$^{-2}$] |
| $\delta$ | gap [m] |
| $\gamma$ | contact angle [rad]; adiabatic index [–] |
| $\kappa$ | adiabatic index [–] |
| $\eta$ | efficiency [–] |
| $\varphi$ | operating angle [rad] |
| $\mu$ | dynamic viscosity [Pa s] |
| $\theta$ | angle [rad] |
| $\rho$ | density [kg m$^{-3}$] |
| $\omega$ | angular velocity [rad s$^{-1}$] |

*Subscripts*

| | |
|---|---|
| * | critical |
| c | cylinder |
| ctfg | centrifugal |
| cv | control volume |
| dcv | discharge control volume |
| disc | discharge condition |
| e | exit |
| endf | endface |
| exp | expander |
| f | friction |
| gen | generated |
| i | inlet, inner |
| isen | isentropic |
| o | outlet, outer |
| orif | orifice |
| p | pressure |
| r | rotor |
| rad | radial clearance |
| rs | rotor shaft |
| s | isentropic |
| scv | suction control volume |
| suct | suction condition |
| t | throat |
| v | vane |
| ve | exposed vane |
| vh | vane head |
| vol | volumetric |
| vs | vane slot |

because the cylinder is allowed to rotate with the rotor, resulting in reduced relative velocities at the rubbing surfaces. This, in turn, reduces the friction loss and maximizes the energy efficiency of the mechanism. There are various design variants of the RV mechanism, following the arrangement of the vane and the driving component (Subiantoro and Ooi, 2010; Teh and Ooi, 2006, 2008).

In the first part of this paper series (Subiantoro and Ooi, in press), the experimental investigation of the RV expander prototype, based on the RV-0 design, has been presented. In this design, the vane is allowed to slide in its slot located at the rotor and to swivel with respect to the cylinder. A schematic diagram of the working principle of the expander mechanism is shown in Fig. 1. The RV-0 mechanism itself has been investigated for compressor applications, both theoretically (Teh and Ooi, 2009b,c; Teh et al., 2009) and experimentally (Teh and Ooi, 2009a). In this paper, mathematical models for the mechanism when used for expander applications are formulated. The models include the geometrical, kinematics, dynamics, thermodynamics, flow through ports and internal leakage models. Whenever possible, they are built based on the methods and steps of the relevant theoretical models previously developed (Fukuta et al., 2009; Jia et al., 2009; Ooi, 2008; Stosic et al., 2003; Teh and Ooi, 2009a, 2009b, c; Teh et al., 2009; Yanagisawa et al., 1984; Yanagisawa and Shimizu, 1985a, b, c; Yang et al., 2009). They are then verified by comparing with the measured data.

## 2. Theoretical models

### 2.1. Geometrical and kinematics models

To begin the formulation of the geometrical and the kinematics models of the expander, the thin vane assumption is made and therefore, the vane thickness and the vane slot volume are assumed to be negligible as compared to the other main dimensions. The models are developed with the geometrical model shown in Fig. 2. Points C and R are the cylinder and the rotor rotation centres, respectively. With the thin vane assumption, the contact between the vane and the vane slot can be assumed to always be at point A, which is the