Dynamic modelling and experimental validation of Coupled Vane Compressor

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Abstract. The newly developed Coupled Vane Compressor (CVC) consists of a unique feature that is a couple of vanes cut through the rotor diametrically. Theoretically, this allows the rotor of any diameter which can accommodate the coupled vanes, to be able to operate CVC. Thus, unlike other rotary compressors, CVC design is mostly free of geometrical constraints imposed on the size of its rotor, making it one of the most compact compressors. Nevertheless, CVC suffers from frictional losses similar to other rotary compressors and therefore, studies of dynamic forces acting in CVC is of great interest. In this paper, dynamic forces acting on its components and frictional losses occurring due to rubbing of various components were studied and analysed. Unlike in other vane compressors, in which frictional losses are mostly due to the rubbing of vane tips against cylinder wall and vane sides against vane slot in the rotor, in CVC, the loss also occurs due to rubbing between the coupled vanes. A CVC prototype of 44 cm³ was developed and the compressor power consumption was experimentally studied in an open-type test circuit using air as an operating fluid. The predicted and measured power consumed by CVC were compared and the predicted results were found to have maximum discrepancy of ±15% with the measured results.

1. Introduction
A new compact rotary vane compressor, namely, Coupled Vane compressor (CVC) was introduced by Ooi and Shakya [1]. As seen from figure 1, a set of two vanes cut diametrically through the rotor. The rotor has rotational motion about its centre, C_r. During the operation, vanes extend radially outward due to centrifugal force and gas pressure forces causing the vane tips to rub against the cylinder wall. The vanes partition the space inside compressor cylinder into various working chambers. The detailed mathematical models describing the working chamber, thermodynamics and mass flow have been studied by Ooi and Shakya [2, 3].

The accurate predictions of the motion and forces acting on various components of a compressor are essential to study its frictional losses and to optimize its design. In various studies by Yanagisawa et al. [4], Ooi and Wong [5] and Ooi [6], rotary vane compressors such as rolling piston compressors are reported to have high frictional losses at the vane tip due to its rubbing against the roller, vane sides with the vane slot and between the roller and eccentric. Ooi [6] reported that frictional losses due to the rubbing of roller against eccentric and against the endface accounted for more than 60% of the total frictional loss. In a study of frictional losses in sliding vane machine by Badr et al. [7] and Aradau and
Costiuc [8], it was found that the highest frictional losses were contributed by vane tip rubbing against cylinder wall.

To mitigate high frictional losses in rotary vane, revolving vane compressor was developed by Teh and Ooi [9]. The frictional losses were reduced by using a rotating cylinder, as opposed to a fixed cylinder. An embodiment of revolving vane compressor was predicted to have the mechanical efficiency of up to 95%. In a swing vane compressor studied by Hu et al. [10], nearly 30% reduction in frictional losses at 3000 r min⁻¹ compared to the same in a sliding vane compressor was predicted. Their design consisted of vane tip connected to cylinder in a vane slot via a hinge joint. Ma et al. [11] conducted a structural optimization of swing vane compressor using complex method and reported an improvement in energy efficiency ratio by 8.5% compared to their base design. However, only three design variables including cylinder radius, rotor radius and the compressor height were selected, and their optimal values tended towards the upper limit of the scope of design variables. By adding a second swing vane, Xu et al. [12] proposed a double swing vane compressor. Their analysis showed lower frictional losses and smoother load fluctuations compared to the same of a swing vane compressor.

In a rotating spool compressor, Bradshaw et al. [13] studied the frictional losses due to spool seal, rubbing of vane tip seals against the cylinder, endface against the cover plate, shearing of fluid at top dead centre region and between vane side and rotor. In their investigation, the design of spool compressor was optimized with reportedly 15% increase in the efficiency. In a study of U-vane compressor by Lim and Ooi [14], the rubbing of vane side with cylinder wall contributed to over 35% of the total frictional loss.

This paper focuses on the study of dynamics of CVC mechanism and analysis of frictional losses due to rubbing of vane tips against cylinder wall, vane sides against vane slot in the rotor and between the vanes.

2. Mathematical models

In this section, the mathematical formulations for the kinematics and dynamics of vanes and rotor are shown in considerable details.

2.1. Kinematics model

Referring to figure 1, the position vectors of the trailing vane tip and leading vane tip with respect to C_r and their derivative with respect to the rotor angle \( \theta_r \) are derived. Their magnitudes are shown in equations (1) - (4).

\[
\begin{align*}
\mathbf{r}(\theta_r) &= \begin{cases} 
R_r & (\theta_{r,st} \geq \theta_r > 360^\circ - \theta_{r,st}) \\
-b \cos \theta_r + \sqrt{-(b \sin \theta_r)^2 + R_c^2} & (\theta_{r,st} \geq \theta_r \\
0 & (\theta_r > 360^\circ - \theta_{r,st})
\end{cases} \\
\frac{d\mathbf{r}(\theta_r)}{d\theta_r} &= \begin{cases} 
R_r & (180^\circ - \theta_{r,st} \leq \theta_r \leq 180^\circ + \theta_{r,st}) \\
b \sin \theta_r + 0.5(-b^2 \sin 2\theta_r)(-(b \sin \theta_r)^2 + R_c^2)^{-0.5} & (180^\circ - \theta_{r,st} \leq \theta_r \leq 180^\circ + \theta_{r,st})
\end{cases} \\
\mathbf{r}(180^\circ + \theta_r) &= \begin{cases} 
-(b \cos(180^\circ + \theta_r) + \sqrt{-(b \sin(180^\circ + \theta_r))^2 + R_c^2}) & (180^\circ - \theta_{r,st} \leq \theta_r \leq 180^\circ + \theta_{r,st})
\end{cases} \\
\frac{d\mathbf{r}(180^\circ + \theta_r)}{d\theta_r} &= \begin{cases} 
0 & (180^\circ - \theta_{r,st} \leq \theta_r \leq 180^\circ + \theta_{r,st})
\end{cases}
\end{align*}
\]
2.2. Vane Dynamics model

The forces acting on a vane can be analysed using a free body diagram. The various forces acting on the free body of the trailing and leading vane at any arbitrary rotor angular position \( \theta \) are shown in figures 2 and 3.

From figures 2 and 3, it is noted that, for CVC to compress the gas without internal leakage through the vane tip, it is required that the vane tip be in sealing contact with the inner cylinder wall. To ensure the vane tips are in sealing contact throughout the operation of the compressor, firstly, the vane tip pressure forces \( F_{t,p1}, F_{t,p2}, F_{t,p3} \) and \( F_{t,p4} \) are required to be smaller than the vane neck pressure forces \( F_{H,p2} \) and \( F_{H,p4} \). At the trailing vane tip, trailing side pressure \( p_t \) is always less than or equal to leading side pressure \( p_l \). This implies that the curved tip area facing the leading side should be made smaller than the curved tip area facing the trailing side. Similarly, at the leading vane tip, trailing side pressure \( p_t \) is always less than leading side pressure \( p_l \). Again, this implies that to minimize the vane tip pressure forces, leading tip curved surface area exposed to \( p_t \) should be smaller than the trailing side curved surface area. Finally, the vane neck pressure forces \( F_{H,p2} \) and \( F_{H,p4} \) should be allowed to be as large as appropriate to ensure that the vane tip remains in contact with the cylinder wall.

The rotation of trailing vane and leading vane about the rotor centre \( C_r \) causes two additional rotational forces to exist for each vane. The first one is centrifugal force \( (F_{cen}) \). And the second one is coriolis force \( (F_{cor}) \) which is due to both rotation and translation of the vanes with respect to \( C_r \).\( F_{cen} \) assists the vane to radially extend out of the rotor slot and towards the inner cylinder wall. Therefore, it is important that vane centre of mass is designed such that the direction of centrifugal force with respect to \( C_r \) is always pointed towards the vane tip to avoid vane retracting from the vane-tip contact.

The pressure forces are computed using chamber pressures from the thermodynamics model and respective force areas. The rotational forces are determined using the vane mass, its centre of mass and operating speed. The ‘unknown’ forces are the contact forces between the vane and other components. They are evaluated by assuming static and dynamic equilibrium at any moment of time.

The rotor provides rotating force, \( F_{N,rot} \), which is the main driving force for the compressor. The vane rotation is assumed to be proportional to the frictional forces, \( F_f, \) and \( F_{f,vn} \). Equation (5) is derived using the force equilibrium on a free body diagram of trailing vane (as shown in figure 2) along \( r(\theta) \).

\[
(F_{t,1} + F_{f,vn} + F_{f,rot}) = F_{r,1} + F_{H,p2} + F_{t,p4} - F_{tp,p1} \cos \left( \frac{\alpha_1}{2} \right) - F_{tp,p2} \cos \left( \frac{\alpha_1}{2} \right) \tag{5}
\]

Equation (6) represents the force balance which are orthogonal to \( r(\theta) \).

\[
(F_{t,1} + F_{N,nn} - F_{N,rot}) = F_{p1} - F_{p2} + F_{tp,p1} \sin \left( \frac{\alpha_1}{2} \right) + F_{tp,p2} \sin \left( \frac{\alpha_1}{2} \right) + F_{cor,1} \tag{6}
\]

Equation (7) can be derived by considering the sum of moments at the centre of mass of trailing vane assuming the anticlockwise rotation of its free body as a positive.

\[
- (F_{t,1} r(\theta) - r_{CM,1}) + F_{N,nn} l_{vn} - r(\theta) + r_{CM,1}) + F_{N,rot} \{R_r - r_{CM,1}\}
= (F_{p2} - F_{p1}) \left[ R_r + \left( \frac{r(\theta) - R_r}{2} \right) - r_{CM,1} \right]
- \left\{ F_{tp,p2} \sin \left( \frac{\alpha_1}{2} \right) + F_{tp,p1} \sin \left( \frac{\alpha_1}{2} \right) \right\} \{r(\theta) - R_r - r_{CM,1}\} \tag{7}
\]

Similarly, equations (8), (9) and (10) are the force balance equations along \( r(180^\circ + \theta_c) \), orthogonal to \( r(180^\circ + \theta) \) and sum of moments at the centre of mass of leading vane respectively.

\[
(F_{l,2} - F_{f,vn} - F_{f,rot}) = F_{r,2} + F_{H,p2} + F_{t,p4} - F_{tp,p3} \cos \left( \frac{\alpha_2}{2} \right) - F_{tp,p4} \cos \left( \frac{\alpha_2}{2} \right) \tag{8}
\]

\[
(F_{l,2} + F_{N,nn} - F_{N,rot}) = F_{p3} - F_{p4} - F_{tp,p3} \sin \left( \frac{\alpha_2}{2} \right) - F_{tp,p4} \sin \left( \frac{\alpha_2}{2} \right) - F_{cor,2} \tag{9}
\]

\[
- (F_{l,1} r(\theta + \pi) - r_{CM,2}) + F_{N,nn} l_{vn} - r(\theta + \pi) + r_{CM,2}) + F_{N,rot} \{R_r - r_{CM,2}\}
= (F_{p4} - F_{p3}) \left[ R_r + \left( \frac{r(\theta + \pi) - R_r}{2} \right) - r_{CM,2} \right]
+ \left\{ F_{tp,p4} \sin \left( \frac{\alpha_2}{2} \right) + F_{tp,p3} \sin \left( \frac{\alpha_2}{2} \right) \right\} \{r(\theta + \pi) - R_r - r_{CM,2}\} \tag{10}
\]
The six unknowns $F_{//,1}$, $F_{//,2}$, $F_{\perp,1}$, $F_{\perp,2}$, $F_{N,\text{rot}}$ and $F_{N,vn}$ are determined by solving six simultaneous equations (5 - 10). Then, the resultant force at the trailing and leading vane tips are determined using equation (11) and (12). $\gamma_1$ and $\gamma_2$ are the angles made by the $F_{tp,1}$ and $F_{tp,2}$ with respect to $r(\theta)$ and $r(180^\circ+\theta_i)$.

$$F_{tp,i} = \sqrt{F_{//,i}^2 + F_{\perp,2}^2}$$

(11)

$$\gamma_i = \tan^{-1}\left(\frac{F_{\perp,1}}{F_{//,1}}\right)$$

(12)

where, $i = 1, 2$

Since the vane tips follow the circular cylinder wall, the force normal to the circular path can be determined using equation (13).

$$F_{N,tp,i} = F_{tp,i} \cos(\gamma_i - \alpha_i)$$

(13)

The frictional forces at the vane tips are proportional to the normal forces derived in equation (13). The frictional forces $F_{f,tp,1}$ and $F_{f,tp,2}$, $F_{f,rot}$, $F_{f,vn}$ can be determined using equations (14), (15) and (16).

$$F_{f,tp,i} = \mu_f F_{tp,i} \cos(\gamma_i - \alpha_i)$$

(14)

$$F_{f,rot} = \mu_f F_{N,\text{rot}}$$

(15)

$$F_{f,vn} = \mu_f F_{N,vn}$$

(16)

2.3. Journal bearing model

The rotor-shaft is supported at its two ends by hydrodynamically lubricated journal bearing. Assuming constant viscosity and incompressible flow of lubricating oil into the converging region of a journal bearing, Hirani et al. [15] developed a model to evaluate the torque required to overcome friction due to fluid shearing (shown in equation 17).

$$T_{b,j} = \left[\frac{\pi \mu_{oil} U_b R_j}{\delta_b R_j} \left(2 + \varepsilon\right) \frac{\delta_v}{\delta_b} \frac{W}{2} \sin \varphi\right] R_j$$

(17)

2.4. Rotor endface friction

Fluid shear at the endfaces of rotor can be modelled assuming Couette flow, constant oil viscosity and clearance gap. The torque required to overcome the fluid shear can be determined using equation (18).

$$T_{enf,j} = \frac{\mu_{oil} \pi \omega_r (R_j^4 - R_{zh}^4)}{2 \delta_{r,enf}}$$

(18)

2.5. Power loss due to friction

To this end, the frictional forces acting on the components of CVC have been derived. The power loss due to friction at the vane tips is shown in equations (19). The power loss due to the friction between
the rotor and vane and between the two vanes is shown in equations (20) and (21) respectively. The power loss at the journal bearing and due to the endface friction can be determined as shown in equations (22) and (23) respectively.

\[
P_{t,p,i}(\theta_r) = \left( F_{f,p,i} \omega_r \right) \left[ \cos \alpha_i \cdot r(\theta_r) + \sin \alpha_i \cdot \left( \frac{dr(\theta_r)}{d\theta_r} \right) \right]
\]

\[
P_{r,vn}(\theta_r) = \left( F_{f,rot} \omega_r \right) \left\{ \frac{dr(\theta_r)}{d\theta_r} + \frac{dr(\theta_r + \pi)}{d\theta_r} \right\}
\]

\[
P_{vn}(\theta_r) = 2 \left( F_{f,vn} \omega_r \right) \left\{ \frac{dr(\theta_r)}{d\theta_r} + \frac{dr(\theta_r + \pi)}{d\theta_r} \right\}
\]

\[
\rho_0(\theta_r) = \sum_{j=1}^{2} \omega_r \cdot T_{b,j}
\]

\[
P_{enf} = \sum_{j=1}^{2} \omega_r \cdot T_{enf,j}
\]

The total power loss in CVC is the sum of all the power losses obtained from equations (19) to (23). The total power loss is obtained in equation (24). The corresponding energy loss due to friction is determined using equation (25). Finally, mechanical efficiency of the compressor can be derived, as shown in equation (26), using the energy lost due to friction and the total indicated work obtained from the pressure-volume curve.

\[
P_{f,total}(\theta_r) = \sum_{i=1}^{2} P_{t,p,i}(\theta_r) + P_{r,vn}(\theta_r) + P_{vn}(\theta_r) + P_{enf}(\theta_r) + P_0(\theta_r)
\]

\[
E_f = \frac{1}{\omega_r} \int_{0}^{540^\circ} P_{f,total}(\theta_r) d\theta_r
\]

\[
\eta_{mech} = \frac{E_{comp}}{E_{comp} + E_f} \times 100\%
\]

3. Results and Discussions

3.1. Simulation results

A simulation program was developed in ForTran for the prediction of performance of CVC and REFPROP [16] was used to calculate thermodynamic properties of the working fluid. The simulation study of CVC is performed using the parameters shown in Table 1.

| Operating condition | Main CVC dimensions |
|---------------------|---------------------|
| Volumetric displacement, \( V_{c,max} \) | Rotor radius, \( R_r \) |
| Operating speed, \( \omega_r \) | Cylinder radius, \( R_c \) |
| Working fluid | Length of compressor, \( l_c \) |
| Evaporating temperature, \( T_{evap} \) | Distance between rotor-to-cylinder centre, \( b \) |
| Condensing temperature, \( T_{cond} \) | Vane thickness, \( l_{vn} \) |
| Compressor inlet temperature, \( T_{in} \) | Vane tip length, \( l_{tip,vn} \) |

| | |
|---|---|
| 44 cm³ | 15.5 mm |
| 3000 r min⁻¹ | 27.5 mm |
| R1234yf | 30 mm |
| 7.2°C | 13 mm |
| 54.4°C | 6 mm |
| 33°C | 12.3 mm |
| | 11 mm |
| | 32.3 mm |

The variation of position vectors \( r(\theta_1) \) and \( r(180° + \theta_1) \), with respect to the rotor centre \( C_r \), are shown up to 180° in figure 4(a), since the compressor is symmetrical about its mid-plane containing the two centres \( C_r \) and \( C_c \) (see figure 1). After 180°, the trailing vane becomes leading vane and vice-versa. This implies that after 180° rotor angle, leading vane assumes the position of trailing vane at 0° and vice-versa with the trailing vane. Until about 30° rotor angle which is half of the sealing arc angle, trailing
vane remains inside the rotor and its vane tip is in contact with the sealing arc, hence there is no radial motion [3]. The same applies to leading vane after 150° rotor angle.

![Figure 4.](image)

(a) Variation of radial lengths over 180° rotor angle (b) Variation of centrifugal force for trailing and leading vane ($F_{cen,1}$ and $F_{cen,2}$)

The variations of centrifugal forces are shown for half a revolution in figure 4(b). For the operation of CVC, a necessary design criterion is that centrifugal force acting on the vane must always be in radially outward direction. It can be seen from the figure 4(b), centrifugal force acting on the vane is positive throughout the revolution. This implies that, for selected dimensions of CVC, centrifugal forces aid the vane tips to be in contact with the inner wall of cylinder.

The working chamber pressure and the respective pressure forces acting on the vane body are shown in figures 5(a) and (b) respectively. The pressure forces acting on the trailing and leading faces of the trailing vane and leading vane ($F_{p1}$, $F_{p2}$, $F_{p3}$ and $F_{p4}$ respectively) are shown in figure 5(c). Similarly, the pressure forces acting on the tips of the vane are shown in figure 5(d). The pressure forces acting at the neck of the vane are shown in figure 5(e). The force area at neck of the vane is designed to be the same as the force area at the rear end of the other vane. Therefore, the pressure force acting on the neck of the trailing vane ($F_{H,p2}$) is of the same magnitude as the force acting on the rear end of the leading vane (see figure 5(e)).

The net reaction forces on the trailing and leading vane tips from the cylinder wall are shown in figure 5(f). The direction of the reaction forces ($F_{//,1}$ and $F_{//,2}$) are antiparallel to the position vectors $r(\theta_r)$ and $r(180° + \theta_r)$. It was found that for the selected design dimension of CVC, $F_{//,1}$ and $F_{//,2}$ were negative when the vane tip receded into the vane slot in the rotor. The negative reaction force implies that the vane tip detaches from cylinder wall. For the trailing vane tip, this occurs at rotor angles $0°$ to half the sealing arc angle, $\theta_{r,st}$, at which the vane tip is inside the rotor slot. For the leading vane tip, this occurs at rotor angles close to $180° - \theta_{r,st}$ at which the vane tip is entering into the vane slot. Upon closer inspection of forces acting on the vane, it was found that this occurred due to two reasons. First, the pressure force at the vane necks ($F_{H,p2}$ and $F_{H,p4}$) were weaker at these positions because of the expansion of the gas exposed at the vane neck at those positions. Secondly, it was assumed that the pressure at sealing arc of CVC was equal to the average of the discharge chamber pressure and the suction chamber pressure. This meant that higher pressure forces were acting on the vane tip to push the vane away from the cylinder wall. This phenomenon can be remedied using either of or combination of two approaches: by changing the length of the vane such that the centrifugal force on the vane is higher which causes larger force to push the vane against cylinder wall. The second way is by modifying the design of sealing arc such that the depth at which the rotor cuts into the cylinder wall ($\delta_{cr}$) is lowered such that net reaction force acting on the vane during the transition from the vane slot to the working chamber is larger. In practice, vane tip detachment from cylinder wall will cause internal leakage and vane chattering.
Figure 5. (a) Various working chamber pressures. (b) Variation of working chamber pressures. (c) Variation of pressure forces. (d) Variation of pressure forces at the trailing vane tip ($F_{tp,p1}$ and $F_{tp,p2}$) and at the leading vane tip ($F_{tp,p3}$ and $F_{tp,p4}$). (e) Variation of the pressure forces at the neck of the trailing ($F_{H,p2}$) and leading vane ($F_{H,p4}$). (f) Variation of resultant force pushing the trailing and leading vane ($F_{//,1}$ and $F_{//,2}$)

After reaction forces were evaluated, normal forces and frictional forces acting between the rubbing components were determined. Figure 6(a) shows frictional force acting between various rubbing components of CVC.

Figure 6. (a) Frictional force at the trailing vane tip ($F_{f,tp1}$) and cylinder wall, leading vane tip and cylinder wall ($F_{f,tp2}$), trailing vane and vane slot in the rotor ($F_{f,rot}$) and between the vanes ($F_{f,vn}$). (b) Power lost per revolution due to the rubbing of CVC components
As shown in figures 2 and 3, a couple of normal forces \( F_{N,rot} \) are assumed to act between the trailing vane and vane slot and leading vane and vane slot, and as such these forces have same magnitude but opposite direction. In figure 6(a), \( F_{f,rot} \) is the frictional force acting between the trailing vane and vane slot and there exists frictional force between the leading vane and vane slot of same magnitude but opposite direction. Similarly, a couple of normal forces \( F_{N,vn} \), which have same magnitude but opposite direction, is also assumed to exist between the trailing and leading vanes. \( F_{f,vn} \) in figure 6(a) is the corresponding frictional force due to the rubbing of rear end of trailing vane against leading vane.

The individual power lost per revolution (figure 6(b)) shows that the most significant power loss occurred at the vane tips. In figure 6(b), the other losses included the fluid frictional losses at the rotor endface, sealing arc, lower and upper bearing. By summing all the individual power losses, the total power loss due to component rubbings in a working cycle of CVC was found to be 353 W. The indicated power from the indicator diagram was 1116 W. Using equation (29), the mechanical efficiency of CVC prototype was determined to be approximately 76%. The mechanical efficiency of CVC can be improved by optimizing the design of vanes.

4. Experiment setup

The detailed description of the test circuit has been presented by Shakya and Ooi [3] and the picture of testbed is shown in figure 7. A CVC prototype with the maximum suction chamber volume of 44 cm\(^3\) was designed and experimentally measured. For simplicity and to save cost, an open-test circuit using air as the working fluid was used. The compressor was powered by ABB 2.2 kW two-pole induction motor and a frequency controller was used to control the operating speed of the compressor. The power input into the compressor was measured using Fluke MDA-510 scope meter. The atmospheric air was directly drawn into the compressor through its suction port and discharged into the discharge tank. A needle at the outlet of the flowmeter was used for the flow control. The uncertainties of various measuring devices used in the testing of CVC are shown in Table 2.

| Table 2. Uncertainties of power measuring Devices |
|-----------------------------------------------|
| Operating frequency, \( \Delta f \)           | ± 2.050 Hz |
| Futek scope meter, \( \Delta I \)            | ± 0.215 A  |

Figure 7. Comparison of the measured and predicted power input

Figure 8. Comparison of the measured and predicted power input

4.1. Results and comparison

The comparison between the predicted and the measured power input to CVC is shown in figure 8. The error bars represent the uncertainties determined for measured power inputs. The frictional coefficient assumed for the prediction was 0.2. For the selected operating conditions, the predicted result was found to have the maximum discrepancy of ±15% with the measured result.
5. Conclusions

CVC’s unique feature is that a couple of vanes cut through the rotor diametrically. Therefore, theoretically, any rotor size which can accommodate the vanes will be able to operate CVC. Thus, allowing it to have the smallest rotor-to-cylinder diameter ratio as compared to all existing rotary vane compressor and hence making it a very compact rotary compressor.

In this paper, the frictional losses due to rubbing components of CVC were analysed. The prediction shows the maximum discrepancy of ±15% with the measurement. The vane tip cylinder friction is the most significant frictional loss. This accounts for nearly 56% of the total frictional loss in CVC. Therefore, for good compressor efficiency of CVC, the optimization of compressor dimensions including vane dimensions are to be carried out.

For further investigation, influence of oil on the frictional losses will be studied and the heat generated due to friction between the rubbing components of CVC will be determined and its effect on the fluid flow and thermodynamics will be analysed.

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Nomenclature

| Subscripts (only abbreviated subscripts are covered) | Greek symbols |
|-----------------------------------------------------|---------------|
| $b$ bearing                                          | $H$ at the neck part of vane |
| $b,f$ bearing friction                               | $j$ journal |
| $cen,1$ centrifugal force at the vane 1 (trailing vane) | $l,vn$ leading vane |
| $cen,2$ centrifugal force at the vane 2 (leading vane) | $N,rot$ normal force on the vane by rotor |
| $cor,1$ coriolis force at the trailing vane          | $N,vn$ normal force at the point of contact between the vanes |
| $cor,2$ coriolis force at the leading vane           | $oil$ oil as the fluid |
| $comp$ compression                                   | $p1$ pressure force at the suction chamber on trailing face of trailing vane side |
| $d$ discharge                                        | $p2$ pressure force at the compression chamber on leading face of trailing vane side |
| $dis$ discharge chamber                              | $p3$ pressure force at the compression chamber on trailing face of leading vane |

Nomenclature

| Nomenclature | Greek symbols |
|--------------|---------------|
| $b$ distance between the rotor centre and the cylinder centre | $a$ contact angle at the vane tip |
| $F$ force | $\gamma$ angle of the resultant force on the vane tip |
| $f$ motor operating frequency | $\varepsilon$ eccentricity ratio |
| $g$ acceleration due to gravity | $\eta$ efficiency |
| $l$ length | $\mu$ frictional coefficient |
| $P$ power | $\theta$ rotation angle |
| $p$ pressure | $\omega$ angular speed ; natural frequency |
| $R$ radius | $[\text{m}]$ |
| $r$ radial coordinate | $[\text{m}]$ |
| $W$ load acting on the bearing | $[\text{N}]$ |

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| $cor,2$ coriolis force at the leading vane           | $oil$ oil as the fluid |
| $comp$ compression                                   | $p1$ pressure force at the suction chamber on trailing face of trailing vane side |
| $d$ discharge                                        | $p2$ pressure force at the compression chamber on leading face of trailing vane side |
| $dis$ discharge chamber                              | $p3$ pressure force at the compression chamber on trailing face of leading vane |
$f$ friction

$p4$ pressure force at the discharge chamber on leading face of leading vane

$f,rot$ frictional force between rotor and vane

$r$ rotor

$f,vn$ frictional force between the vanes

$enf,vn$ vane endface

$r,enf$ endface of the rotor

$r,st$ at the start of the suction with respect to the rotor centre

$gap$ gap

$s$ suction

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