Full Length Research Paper

Optimization of R134a cross vane expander compressor refrigeration system oriented to COP

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Cross vane expander compressor (CVEC) is a newly invented expander-compressor combined unit, where it is introduced to replace the compressor and the expansion valve in traditional refrigeration system. The mathematical model of CVEC has been developed to examine its performance and it was found that the energy consumption of a conventional refrigeration system was reduced by as much as 18%. It is believed that this result can be further improved by optimizing the device. In this paper, the coefficient of performance (COP) of CVEC has been optimized under predetermined operational conditions. Several main design parameters of CVEC were selected to be the variables and the optimization was done with theoretical model in a simulation program. The theoretical model consists of geometrical model, dynamic model, heat transfer model and valve dynamics model. Complex optimization method, which is a constrained, direct search and multi-variables method, was used in the study. As a result, the optimization study suggested that with an appropriate combination of design parameters, a large COP improvement in CVEC R134a refrigeration system is possible. With a reduction of 20% in the shaft radius and 17% in inner cylinder radius, and the increase of around 16% in ports diameter, an improvement of 58% in COP can be achieved.

Key words: Coefficient of performance (COP), cross vane expander-compressor, simulation, optimization.

INTRODUCTION

Cross Vane Expander Compressor (CVEC) (patent number: WO2013162477) was invented by Yap et al. (2013) from Nanyang Technological University. Unlike most of the expander compressor units which are made up by two separated machine, in CVEC, both the compression and expansion process is done within a cylinder (Yap et al., 2014), thus its size is more compact compared to other compressors of the same capacity, such as reciprocating compressor (Baek et al., 2002), the scroll compressor (Kakuda et al., 2009), the rotary screw compressor (Smith et al., 1994), the revolving vane compressor (Subiantoro and Ooi, 2009), and the rolling-piston compressor (Li et al., 2009).

As there is energy recovered from the expansion...
process, CVEC is believed to have more energy saving, more efficient and environmentally friendly. Figure 1 shows the isometric view of CVEC, from which, some parts are shared between the compressor and expander, and most of the parts are cylindrical. Thus its manufacturing cost is expected to be similar to a single compressor.

Figure 2 shows the assembly of major components of CVEC. The compressor is between the outer cylinder and inner cylinder, whereas the expander is within the inner cylinder. The rotation of vane is driven by a drive shaft which is coupled with a prime mover. During operation, the vane revolves, which in turn rotates the cylinders, causes the volumes of all the chambers of the cylinders vary and from which, the compression and expansion happens. CVEC has attracted much attention from the industries and research centres due to the following advantages:

1. Fewer number of parts to be fabricated and manufactured, thus less material used lead to more environmentally friendly.
2. Lower power consumption and more energy saving due to the work recovered from expansion process.
3. Smaller physical size, more compact and space saving.
4. Simplicity in implementation as it is an integrated machine.

However, with the current computer technology, a design optimization can be carried out for a higher efficiency and higher performance CVEC. A suitable optimization technique as well as the design constraints and the objective function is required for the optimization of CVEC.

MATHEMATICAL MODEL AND OPTIMIZATION ALGORITHM

A few sets of coupled mathematical equations have been developed to express the operating cycle of CVEC. This mathematical model describes number of complicated phenomenon in the compressor and the expander of CVEC.

Geometrical model

Figures 3 and 4 show the schematic of the CVEC. The volume of the working chambers of both compressor and expander vary when the vane is rotating. Suction chamber volume $V_{escv}$ and discharge chamber volume $V_{edcv}$ of the expander are further derived as shown in (1) and (2):

$$V_{escv} = \frac{1}{2}[r_{ic}^2 \theta_2 - r_i^2 \theta_1 - (l_i + r_s)(r_{ic} - r_s)sin\theta_1 - l_i w_c] \epsilon$$  \hspace{1cm} (1)
Due to the swiveling motion of the split bush about the hinge joint, the Split Bush Frictional Loss $P_{sb}$ (8) is defined by Equation 7:

$$P_{sb} = |\Delta \omega| r_{sb} h_{sb} \sqrt{F_{sb}^2 + F_{sbn}^2}$$

where $r_{sb}$ is the radius of the split bush, $\mu_{sb}$ is the coefficient of friction between split bush and the hinge joint, $\Delta \omega$ is the relative angular velocity with respect to the hinge joint, $F_{sb}$ is the contact force between the split bush and the hinge joint in the direction opposite to vane side friction, and $F_{sbn}$ is the contact force between the split bush and the hinge joint in the direction normal to the vane.

Both $F_{sb}$ and $F_{sbn}$ can be referred from Figure 10 and are defined as Equations 8 and 9 (Yap, 2012):

$$F_{sb} = \mu_{es} \left( \frac{I_{ic} \omega_2}{r_{ic} + \frac{l_{ic}}{2}} \right) \cos \gamma$$

$$F_{sbn} = \left( \frac{I_{ic} \omega_2}{r_{ic} + \frac{l_{ic}}{2}} \right) \cos \gamma$$

where $I_{ic}$ is the moment of inertia of the inner cylinder.

Vane side friction

The Vane side frictional loss $P_{sv}$ (8, 9) which occurs when the vane slides along the slot of the split bush during operation, is defined as (10):

$$P_{sv} = \mu_{es} \left( \frac{I_{ic} \omega_2}{r_{ic} + \frac{l_{ic}}{2}} \right) \cos \gamma \frac{d\omega}{dt}$$

where $\mu_{sv}$ is the coefficient of friction between the vane sides and the slot of the split bush and $\frac{d\omega}{dt}$ is the sliding velocity of the vane (m/s).

Endface friction

Endface friction occurs when there is relative motion between two contact surfaces during operation. In CVEC, endface frictional loss between upper endface of the inner cylinder and outer cylinder

Thermodynamics model

Due to the varying volume of working chambers, the fluid properties such as pressure, temperature, and density vary. These variations of thermodynamics properties can be simulated based on the Law of Conservation of Energy presented by Moran and Shapiro (Moran and Shapiro, 2008), which is given by Equation 5:

$$\frac{dE_{cv}}{dt} = Q_{cv} - W_{cv} + \sum m_i \left( h_i + \frac{v_i^2}{2} + gz_i \right) - \sum m_o \left( h_o + \frac{v_o^2}{2} + gz_o \right)$$

where $E_{cv}$ is the total energy of the control volume, $Q_{cv}$ is the net heat transfer into the control volume, $W_{cv}$ is the net work done by the control volume, $m_i$ is the mass of fluid flowing into the control volume, $m_o$ is the mass of fluid flowing out of the control volume, $h_i$ is the specific enthalpy of the fluid flowing into the control volume, $h_o$ is the specific enthalpy of the fluid flowing out of the control volume, $v_i$ is the velocity of the fluid flowing into the control volume, $v_o$ is the velocity of the fluid flowing out of the control volume, $z_i$ is the height of the fluid flowing into the control volume, $z_o$ is the height of the fluid flowing out of the control volume, $g$ is the acceleration of free fall, $t$ is the time.

The second term on the right hand side of the equation represents work done due to volumetric change, which can be expressed as:

$$\frac{dV_{cv}}{dt} = p_{cv} \frac{dV_{cv}}{dt}$$

Where $p_{cv}$ is the pressure of the control volume and $V_{cv}$ is the volume of the control volume.

Frictional losses

Split bush friction

Suction chamber volume $V_{scc}$, and discharge chamber volume $V_{dcv}$ are derived as shown (3) and (4) (Yap, 2012):

$$V_{scc} = \frac{1}{2} [r_{ic}^2 (\theta_2 - \pi) + (\pi - \theta_1) (r_{ic} + t_{ic})^2 + (l_{1+ic} + r_i) (r_{ic} - r_i) \sin \theta_1 - (l_\omega - l_{1+ic}) w_{ic}] |_{eps}$$

$$V_{dcv} = \left[ \pi (r_{oc}^2 - (r_{ic} + t_{ic})^2) - (l_\omega - l_{1+ic}) w_{ic} \right] |_{eps} - \frac{1}{2} [r_{oc}^2 (\theta_1 - \pi) + (\pi - \theta_1) (r_{ic} + t_{ic})^2 + (l_{1+ic} + r_i) (r_{ic} - r_i) \sin \theta_1 - (l_\omega - l_{1+ic}) w_{ic}] |_{eps}$$

where $l_{1+ic}$ is the length from point A to point D in Figure 5, $\theta_1$ is the rotational angle as shown in Figure 4, $r_{ic}$ is the inner cylinder radius and $r_i$ is the shaft radius.

Endface friction occurs when there is relative motion between two contact surfaces during operation. In CVEC, endface frictional loss between upper endface of the inner cylinder and outer cylinder...
(\(P_{ic,oc}\)), endface frictional loss between lower endface of the inner cylinder and the surface of the lower bearing \((P_{ic,lb})\), and endface frictional loss between lower endface of the outer cylinder and the surface of the lower bearing \((P_{oc,lb})\) are defined in (11), (12) and (13) respectively, which are formulated similar to RV mechanism (Equation 11):

\[
P_{ic,oc} = \frac{\mu u_{ef}^2}{2\delta_{ic}} ((r_{ic} + t_{oc})^2 - r_{ic}^2) + \frac{\mu u_{ef}^2}{2\delta_{ic}} ((t_{ic} + t_{lb})^4 - r_{ic}^4)
\]

(11)

\[
P_{ic,lb} = \frac{\mu u_{ef}^2}{2\delta_{ic}} ((r_{ic} + t_{lb})^2 - r_{ic}^2)
\]

(12)

\[
P_{oc,lb} = \frac{\mu u_{ef}^2}{2\delta_{ic}} ((t_{oc} + t_{lb})^2 - r_{ic}^2)
\]

(13)

Where \(\delta_{ic}\) is the lower endface clearance, \(\delta_{lb}\) is the upper endface clearance and \(\mu\) is the dynamics viscosity of the lubricant.

**Instantaneous power**

In CVEC, the actual instantaneous power consists of compression power \((P_{com})\), expander power \((P_{exp})\) and power due to inertia \((P_{i})\) (Equations 8, 9). Each of these powers is defined as Equations 14, 15 and 16:

\[
P_{com} = (p_{edcv} - p_{escv}) (t_i - l_{i+1}) \omega_1 \left( \frac{l_1}{2} + l_i \right) \omega_1
\]

(14)

\[
P_{exp} = (p_{escv} - p_{edcv}) l_1 \omega_1 \left( \frac{l_1}{2} + l_i \right) \omega_1
\]

(15)

\[
P_{i} = l_i \omega_2 \omega_2
\]

(16)

Where \(p_{edcv}\) is the pressure of compression chamber of the compressor and \(p_{escv}\) is the pressure of suction chamber of the compressor, \(p_{escv}\) is the pressure of suction chamber of expander and \(p_{edcv}\) is the pressure of discharge chamber of expander, \(\omega_2\) is the angular acceleration of the inner cylinder. Based on the above frictional losses, the total frictional power loss is defined as Equation 17:

\[
\sum P_{loss} = P_{va} + P_{eb} + P_{ic,lb} + P_{oc,lb} + P_{ic,oc} + P_{value}
\]

(17)

where \(P_{value}\) is the suction and discharge loss between the ideal and actual case. From all the defined power losses, the instantaneous power is defined as Equation 18:

\[
P_{instant} = P_{com} - P_{exp} + P_{i} + \sum P_{loss}
\]

(18)

**Coefficient of performance**

As the compression work in CVEC is supported by both the prime mover and the work recovered from the expansion, the Coefficient of Performance (COP) is increased due to lesser work required by the compressor. In fact, the expander is considered as a near isentropic device and thus the discharged fluid from the expander has a lower enthalpy as compared to the discharged fluid from the expansion valve. This results in larger cooling capacity which can be as much as 15% (12), and the COP is defined as Equation 19:

\[
COP = \frac{\Delta h_{ns}}{W_{motor} - W_{exp}}
\]

(19)

where \(\Delta h_{ns}\) is the difference in enthalpy between the discharge fluids from the expander and expansion valve, \(W_{motor}\) is the required motor work without the expander and \(W_{exp}\) is the work recovered from expander.

The optimization study of this paper involved the linking of the theoretical simulation model with an optimization algorithm. This allows a set of combination of design parameters that produces optimum performance of CVEC at predetermined operational conditions and within the design constraints. In this study, Complex Optimization Method which was developed by Box (1965) was used implemented.

The optimization study (14) can be mathematically represented by:

1. To maximize/minimize objective function: \(F(x) = f(x_1, x_2, ..., x_N)\) with explicit, geometrical and implicit constraints:

\[
L_E(x)_i \leq E(x)_i \leq H_E(x)_i, \quad i = 1, 2, ..., N
\]

(20a)

\[
L_G(x)_j \leq G(x)_j \leq H_G(x)_j, \quad j = 1, 2, ..., M
\]

(20b)

\[
L_I(x)_k \leq I(x)_k \leq H_I(x)_k, \quad k = 1, 2, ..., L
\]

(20c)

where \(E, G\) and \(I\) stand for explicit, geometrical and implicit constraints, \(L\) and \(H\) stand for the lower and higher limit, \(i, j\) and \(k\) are the number of explicit, geometrical and implicit constraints respectively.

2. A starting point that satisfies all constraints is picked. K-1 additional points are generated from pseudo-random numbers and constraints of each independent variable:

\[
x_{i+1} = L_E(x)_i + r_{ij} \times [H_E(x)_i - L_E(x)_i]
\]

(21)

where the additional points generated together with the starting point are called the original complex, \(N\) is the number of explicit free variable constraints and \(K = N + 1\). \(L_E(x)_i\) and \(H_E(x)_i\) are the lower and higher limit of the free variables, \(r_{ij}\) is the pseudo-random number between 0 and 1.

3. The selected points satisfy the explicit constraints of the free variables but may violate other constraints. If either geometric or implicit constraint is violated, the selected point will be moved one half of the distance to the centroid of the remaining point:

\[
x_{i+1}^{(new)} = \frac{x_{i+1}^{(old)} + x_i}{2}, \quad i = 1, 2, ..., N
\]

(22)

with the centroid \(\bar{x}_{ic}\) defined as

\[
\bar{x}_{ic} = \frac{1}{K-1} \sum_{j=1}^{K} x_{i+j} - x_{i+1}^{(old)}, \quad i = 1, 2, ..., N
\]

(23)

The process is repeated until no more constraints are violated.

4. Evaluate the objective function at each point and the point that having the worst objective function value is reflected by reflection factor, \(\alpha = 1.3\), which is suggested by M.J. Box (Box, 1965) along the line which links the replaced point and the centroid. The new point is:

\[
x_{i+1}^{(new)} = (1 + \alpha)\bar{x}_{ic} - \alpha x_{i+1}^{(old)}
\]

(24)
Maximize the COP of CVEC refrigeration

\[
\begin{align*}
\text{Maximize} & \quad f(x) < s, \quad x < i, \quad i = 1, 2, \ldots, 6.
\end{align*}
\]

Subjected to constraints:

\[
\begin{align*}
x_i(t) & < x(i) < x_b(i), \quad i = 1, 2, \ldots, 6.
\end{align*}
\]

The objective function in (27) is defined the same as COP. Since the variations of geometrical design parameters affect the power consumption and the power losses, the objective function is dependent on several design parameters. By direct searching each of the design parameters within their constraints, a set of design parameters can be found that produces the optimum objective function.

In (28), \(x(i), i = 1, 2, \ldots, 6\) are design independent variables \(r_s, r_{ic}, d_{h1}, d_{h2}, d_{h3}\) and \(d_{h4}\). Here, \(d_{h1}, d_{h2}, d_{h3}\) and \(d_{h4}\) are expander suction port diameter, expander discharge port diameter, compressor suction port diameter and compressor discharge port diameter respectively. \(x_i(t)\) and \(x_b(i)\) indicate the lower limit and upper limit of the \(i\)th design parameter. There are implicit geometrical variables such as \(l_{eps}, t_{ic}\), and \(r_{oc}\), which are dependent on the design parameters \(x(i)\). The constraints of these implicit geometrical variables are also fixed to ensure a certain capacity of CVEC.

The values of constraints of each variables mentioned above are given in Tables 1 and 2 so that the dimensions of the final design are within the given range. As \(t_{ic}\) and \(r_{oc}\) are preferred to be as small as possible, only upper limit is given.

### RESULTS AND DISCUSSION

The operational conditions in this optimization study are specified in Table 3. The optimization search was implemented on a HP Prodesk with an INTEL CORE i7 processor and it took 25 hours to complete the search. The simulation program was done using MATLAB (Chapra and Canale, 2010) and the thermodynamics properties of refrigerant were extracted from NIST database using REFPROP (Lemmon et al., 2010). The following assumptions are made during the optimization study.

1. Adiabatic process,
2. Internal leakage is not considered.

### Table 1. Constraints of independent variables used in the optimization of CVEC.

<table>
<thead>
<tr>
<th>S/N</th>
<th>Lower limit</th>
<th>Design variables</th>
<th>Upper limit</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>8.00</td>
<td>(r_s)</td>
<td>12.00</td>
<td>mm</td>
</tr>
<tr>
<td>2</td>
<td>10.00</td>
<td>(r_{ic})</td>
<td>14.00</td>
<td>mm</td>
</tr>
<tr>
<td>3</td>
<td>10.00</td>
<td>(d_{h1})</td>
<td>14.00</td>
<td>mm</td>
</tr>
<tr>
<td>4</td>
<td>10.00</td>
<td>(d_{h2})</td>
<td>14.00</td>
<td>mm</td>
</tr>
<tr>
<td>5</td>
<td>11.00</td>
<td>(d_{h3})</td>
<td>15.00</td>
<td>mm</td>
</tr>
<tr>
<td>6</td>
<td>11.00</td>
<td>(d_{h4})</td>
<td>15.00</td>
<td>mm</td>
</tr>
</tbody>
</table>

### Table 2. Constraints of geometrical variables used in the optimization of CVEC.

<table>
<thead>
<tr>
<th>S/N</th>
<th>Lower limit</th>
<th>Geometrical variables</th>
<th>Upper limit</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>30.00</td>
<td>(l_{eps})</td>
<td>120.00</td>
<td>mm</td>
</tr>
<tr>
<td>2</td>
<td>35.00</td>
<td>(t_{ic})</td>
<td>35.00</td>
<td>mm</td>
</tr>
<tr>
<td>3</td>
<td></td>
<td>(r_{oc})</td>
<td>50.00</td>
<td>Mm</td>
</tr>
</tbody>
</table>

The objective function will be maximized during the optimization search, Ooi (2005) has calculated the centroid by weighting each point according to their objective function value. In this optimization study, a method which is similar to the weighted arithmetic mean has been proposed by the author to calculate the centroid of the remaining points. The following equation is used:

\[
\bar{x}_{ic} = \frac{\sum_{i=1}^{N} x_i \left( \frac{f_i - f_{worst}}{f_{best} - f_{worst}} \right)}{\sum_{i=1}^{N} \left( \frac{f_i - f_{worst}}{f_{best} - f_{worst}} \right)} \quad i = 1, 2, \ldots, N
\]

5. If an explicit constraint is violated after reflection, the point is moved inside its constraint boundary by a factor \(\beta = 0.0001\):

\[
x_{i,j}^{\text{new}} = H_{E}(x) - \beta \quad \text{if higher boundary is violated}
\]

\[
x_{i,j}^{\text{new}} = L_{E}(x) + \beta \quad \text{if lower boundary is violated}
\]

6. The objective function value is checked for any improvement and convergence, if the difference between best objective function value and worst objective function value that satisfies all the constraints lies within a tolerance, the point of the best objective function value will be selected.

This optimization procedure is repeated with few set of initial variables to find several local optimum points. The local optimum points are then compared to find the global optimum points and they will be the optimized design geometrical variables of CVEC.

The objective is to maximize the COP of CVEC refrigeration system, where the objective function will be maximized during the optimization search. The objective function (COP) can be expressed mathematically as:

\[
F_{obj} = f(r_s, r_{ic}, d_{h1}, d_{h2}, d_{h3}, d_{h4})
\]
Table 3. Operating specification of optimization study.

<table>
<thead>
<tr>
<th>Operating specifications</th>
<th></th>
</tr>
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<tbody>
<tr>
<td>Working fluid</td>
<td>R-134a</td>
</tr>
<tr>
<td>Degree of superheated</td>
<td>27.8°C</td>
</tr>
<tr>
<td>Saturated temperature of condenser</td>
<td>54.4°C</td>
</tr>
<tr>
<td>Saturated temperature of evaporator</td>
<td>7.2°C</td>
</tr>
<tr>
<td>Operational speed, ( \omega_1 )</td>
<td>3000 rpm</td>
</tr>
<tr>
<td>Dynamic viscosity of lubricant</td>
<td>0.0034 Pa·s</td>
</tr>
<tr>
<td>Cooling load</td>
<td>5275 W</td>
</tr>
</tbody>
</table>

Table 4. Value of design parameters after optimization.

<table>
<thead>
<tr>
<th>S/N</th>
<th>Design variables</th>
<th>Original value (mm)</th>
<th>Optimized value (mm)</th>
<th>Change (%)*</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>( r_s )</td>
<td>10.00</td>
<td>8.00</td>
<td>-20.00%</td>
</tr>
<tr>
<td>2</td>
<td>( r_{ic} )</td>
<td>12.00</td>
<td>10.00</td>
<td>-16.67%</td>
</tr>
<tr>
<td>3</td>
<td>( d_{h1} )</td>
<td>12.00</td>
<td>14.00</td>
<td>16.67%</td>
</tr>
<tr>
<td>4</td>
<td>( d_{h2} )</td>
<td>12.00</td>
<td>14.00</td>
<td>16.67%</td>
</tr>
<tr>
<td>5</td>
<td>( d_{h3} )</td>
<td>14.00</td>
<td>14.52</td>
<td>3.71%</td>
</tr>
<tr>
<td>6</td>
<td>( d_{h4} )</td>
<td>14.00</td>
<td>14.99</td>
<td>7.07%</td>
</tr>
</tbody>
</table>

*Change (%) = (optimized value – original value)/(original value x 100%)

The changes of the main design parameters are shown in Table 4. It is obviously that the shaft radius has the most significant change, followed by inner cylinder radius, expander suction port diameter and expander discharge port diameter.

With all these changes in main design parameters, which in turn affect the power losses of CVEC, a maximized objective function value has been found within the given design parameters constraints. Figure 6 shows the variation of objective function when the optimization search progresses. The objective function, which is the COP, has been increased by 58% after the optimization study, which implies a lower power consumption as the cooling load is fixed in this study. Based on the improvement in objective function, it also indicates a significant reduction in the total power loss of around half of its initial value, which is shown in Figure 7. Figure 8 to 14 show various frictional power losses that were considered in the mathematical model of CVEC. From the results, it is found that all the frictional losses were decreased after the optimization search, with reduction ranging from 20% to 60% in individual frictional loss. In addition, the maximum contributor in the reduction of the total power loss was the frictional power loss between shaft and bearing, which is 58% as shown in Figure 8. The reduction was caused mainly by the reduction in shaft radius, where the reasons can be traced back to the definition of the definition of power losses as shown in Equations 7 to 16.

From the definition of all the frictional power losses in session 2 and the above results, the reduction of shaft radius and inner cylinder radius has effectively reduced the frictional power losses whereas the increase in port diameter is believed to have improve the valve loss. The valve loss is actually the additional work done required to keep up the working fluid flow with the operating speed. Figures 15 and 16 show the variation of compressor power and expander power during optimization search.
The power required by the compressor was decreased by 1% and the power recovered from the expander was increased by 22%, which indicates a lower power consumption and more power recovery in CVEC.

From the optimization study, a narrower and taller CVEC is suggested that gives better performance, where the length of CVEC \( l_{\text{eps}} \) increased by 22%.

**Conclusion**

The optimization study verified that a compressor design with mathematical model can be implemented a design optimization algorithm. A set of main design parameters was suggested that produces maximum CVEC performance based on the given constraints and operation conditions. From the optimization study, it is suggested that with a proper combination main design parameters, a 50% total frictional power loss can be achieved which gives a 58% COP improvement in CVEC.

**CONFLICT OF INTERESTS**

The authors have not declared any conflict of interests.

**ACKNOWLEDGEMENTS**

The authors express their deepest appreciation and
Figure 11. Variation of split bush power loss during optimization search.

Figure 12. Variation of endface friction between inner cylinder and lower bearing during optimization search.

Figure 13. Variation of endface friction between outer cylinder and lower bearing during optimization search.

Figure 14. Variation of endface friction between inner cylinder and outer cylinder during optimization search.

Figure 15. Variation of compressor power during optimization search.

Figure 16. Variation of expander power during optimization search.
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REFERENCES


