Research on the Performance of a High Pressure 5.3MPa Twin Screw Compressor(1161)

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Content

● Introduction of high pressure twin screw compressor
● Development of the semi-empirical model
● Experimental facility setup
● Results verification and discussion
● Conclusion
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Introduction of high pressure twin screw compressor

Background

- Development in low temperature refrigeration (-35°C and lower)
- Employment of nature refrigerant
- Requirement of large capacity
Specific requirement for CO2 compressor

- Specially high operating pressure
- High loading capacity for moving parts
- Internal leakage control
- Manufacturing difficulty for component
- High discharge temperature
Introduction of high pressure twin screw compressor

- Application of subcritical CO₂ refrigeration system
Operating condition

- Condensing temperature: -15 to 15 °C
- Evaporating temperature: -54 to -15 °C.
- Operating pressure: 5.3MPa
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Approaches for twin screw compressor performance prediction:

- CFD modeling
- Working process simulation
- Conventional semi-empirical model

Drawbacks:

- Complex
- Time consuming
Specific consideration of high pressure compressor

- Inter-stage leakage
- Heat transfer with oil
- Losses in compressing process
Development of the semi-empirical model

- 1-2: suction heating
- 2-3: refrigerant mixing
- 3-4: 1st stage compressing
- 4-5: 2nd stage compressing
- 5-6: 3rd stage compressing
- 6-7: discharge process
- 7-8: heat transfer
Development of the semi-empirical model

- **Refrigerant heat transfer**
  
  \[ Q = f \frac{am}{2\pi n}(t_1 - t_2) \quad f \propto \left(\frac{t_1 - t_2}{L}\right)^{\frac{1}{4}} \]

- **Leakage**
  
  \[ \dot{m}_{leak} = \frac{b}{v} \sqrt{p_1 - p_2} \]

- **Viscous friction loss**
  
  \[ P_{loss} = c\mu_{oil}V_g n^2 \]

- **Heat transfer with oil**
  
  \[ Q_{oil} = \frac{d}{n} m_{oil}(t_{dis} - t_{oil})^{\frac{5}{4}} \]
Development of the semi-empirical model

● 6 parameters calibrated:
  
  a --- suction heating degree
  
  \[ b_{4,5,6} \] --- refrigerant leakage of process 4,5,6
  
  c --- viscous friction power loss
  
  d --- transmission loss

● Identification of parameters

\[
err = \sqrt{\sum \left( \frac{\eta_{sl}^a}{\eta_{a}^{test}} - 1 \right)^2 + \sum \left( \frac{\eta_{sl}^v}{\eta_{v}^{test}} - 1 \right)^2 + \sum \left( \frac{i_{sl}^{test}}{i_{dis}^{test}} - 1 \right)^2}
\]
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Experimental facility setup

Experimental facility
### Experimental facility setup

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lobes of the male rotor</td>
<td>5</td>
</tr>
<tr>
<td>Lobes of the female rotor</td>
<td>7</td>
</tr>
<tr>
<td>Outer diameter of the male rotor, mm</td>
<td>189.5</td>
</tr>
<tr>
<td>Centre distance between the male and female rotors, mm</td>
<td>155</td>
</tr>
<tr>
<td>Length of the rotors, mm</td>
<td>225</td>
</tr>
<tr>
<td>Theoretical volume of gas discharged per round, cm³</td>
<td>3399.85</td>
</tr>
<tr>
<td>Theoretical flow rate of gas discharged, m³·h⁻¹</td>
<td>603.84</td>
</tr>
</tbody>
</table>
Experimental facility setup

Experiment content:

- Stability test
- Vibration and noise test
- Capacity test
- Power consumption test
- Oil cooler test
- Extreme operating condition test
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Results verification and discussion

Validation of model

Volumetric efficiency

- Maximum error: 5%
- Difference of efficiency: 0.2
- Caused by severe leakage
Results verification and discussion

Isentropic efficiency

- Maximum error: 5%
- Difference of efficiency: 0.25
- Caused by internal leakage and suction heating
Results verification and discussion

COP

- Maximum error: 5%
- Difference of COP: 2.2
Results verification and discussion

Volumetric efficiency

- Highest efficiency: 0.92
- Difference of efficiency: 0.2
- Increase with lower compressing ratio
Results verification and discussion

**Isentropic efficiency**

- Highest efficiency: 0.72
- Difference of efficiency: 0.26
- Caused by over/owing-compression
Results verification and discussion

COP

- Highest COP: 6.6
- Difference of efficiency: 2.2
- Caused by increase in both the theoretical COP and isentropic efficiency
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Conclusion

In this paper, a novel semi-empirical model and experimental plant are developed for a high pressure 5.3MPa twin screw compressor.

- The high pressure CO2 twin screw compressor is guaranteed stability and reliability.
- The accuracy of the model could be guaranteed, with an maximum error of 5%.
- The COP ranges from 0.9 to 6.6 according to the operating condition, which decrease with the increase of compressing ratio.
Conclusion

- The adiabatic efficiency will first increase and then decrease the increase of compressing ratio. It may contributes to the over/owing-compression.

- The volumetric efficiency may monotonously increase with increase of compressing ratio. It is caused by the severe leakage and suction heating.
Thanks!