A Methodology for Characterization of Vapor-injection Compressors

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Introduction
In Europe single-stage compressors are characterized using Standard UNE-EN 13771-1 (2003).

**Working conditions:**
- Suction pressure
- Discharge pressure
- Inlet superheat

**Measured variables:**
- Power input
- Refrigerant mass flow rate

The standard provides several procedures for testing compressors:

*Example: Calorimeter with secondary fluid at the suction*

*Source: UNE-EN 13771-1 (2003)*
For single-stage compressors, manufacturers provide AHRI polynomials (AHRI Standard 540) in order to estimate the mass flow rate and the power input of the compressors when they operate in different conditions of the catalog data.

\[ X = C_1 + C_2 \cdot (t_s) + C_3 \cdot t_D + C_4 \cdot (t_s^2) + C_5 \cdot (t_s \cdot t_D) + C_6 \cdot (t_D^2) + C_7 \cdot (t_s^3) + C_8 \cdot (t_D \cdot t_s^2) + C_9 \cdot (t_s \cdot t_D^2) + C_{10} \cdot (t_D^3) \]

Where:

- \( C_1 \) through \( C_{10} \) = Regression coefficients provided by the manufacturer
- \( t_D \) = Discharge dew point temperature, °C
- \( t_s \) = Suction dew point temperature, °C
- \( X \) = Individual Published Ratings: Power Input, W, W
  - Refrigerant Mass Flow Rate, kg/s
  - Refrigerating Capacity, W

With this AHRI polynomial the compressor performance can be determined in any working point.
Introduction

Motivation

- For vapor-injection compressors, the characterization process is more complex.
- There is not a public standard for characterization (only there is a pre-standard 13771:2015)
- Two additional degrees of freedom:
  - Intermediate pressure ($P_{int}$)
  - Injecting Superheat ($Sh_{inj}$)

- An additional variable for measuring:
  - Injection mass flow rate ($m_{inj}$)
- For a given test matrix, the number of experimental points increases because $P_{int}$ can take several values for each working conditions ($Te$, $Tc$, $Sh$)
Introduction

Motivation

- Fixed the intermediate superheat, the intermediate conditions (\(P_{\text{int}}\) and \(\dot{m}_{\text{inj}}\)) depend critically on the way in which the injection is performed (economizer, flash tank, liquid injection, etc.).

- Once the injection procedure is fixed not all the intermediate conditions are independent.
Several compressor manufacturers define the economizer size by setting the temperature approach in the economizer. For all operating points, the temperature approach is the same (5 K).

This do not correspond to any real physical system but for the nominal point. T₇ is the bubble temperature of the intermediate pressure.

A more real approach could be to fix the UA value for the nominal point. Which would be the compressor behavior if another heat exchanger or another injection method is selected? 

Not general and not dependent intrinsically only in the compressor.
Is it possible to characterize the intermediate conditions regardless of the injection process used and without performing a huge amount of tests?
Proposed solution:

In order to characterize vapor-injection compressors independently of the injection procedure, a relation between the injection mass flow and the intermediate pressure must be defined.

This correlation will be analogous to an AHRI polynomial of single-stage compressors.

$$\dot{m}_{\text{inj}} = f(\dot{m}_{\text{evap}}, P_{\text{evap}}, SH, P_{\text{cond}}, P_{\text{inj}}, SH_{\text{inj}})$$

A characterization of this compressor technology could be thought in terms of a function for that relation.

We can use this correlation in a simple model of a the cycle in order to predict the compressor performance operating in any working points and any injection system.
Objective of the work

- Establish a characterization methodology for vapor-injection compressors which depend on only in compressor characteristics.

Calorimetric bench for testing injection compressor.

Determination of the intermediate conditions correlation based on the previous tests:

\[
\dot{m}_{\text{inj}} = f(\dot{m}_{\text{evap}}, P_{\text{evap}}, SH, P_{\text{cond}}, P_{\text{inj}}, SH_{\text{inj}})
\]

Evaluation of the obtained results in a heat pump prototype.
Experimental Setup
The experimental setup consists of a typical calorimetric bench with the addition of the injection line. The intermediate pressure is controlled by an electronic expansion valve and the injection superheat is fixed by the water-glycol temperature throughout a heat exchanger.

The injection line is separate to the evaporator line in order to control independently the intermediate pressure and the injection superheat.
Experimental Setup

The 23rd International Compressor Engineering Conference at Purdue

\[ \dot{m}_{\text{evap}} = \dot{m}_{\text{cond}} - \dot{m}_{\text{inj}} \]
Experimental Setup

Calorimetric bench
Experimental Setup

Heat pump prototype

Diagram shows a heat pump prototype with labels for different components:
- Condenser
- Economizer
- Evaporator
- Climatic Chamber

The diagram includes various symbols for pressure (P) and temperature (T) measurements, as well as flow (m) indicators.
Test Matrix
Test Matrix

- Swept volume 17.8 m$^3$/h
- Refrigerant R407C
- 2900 rpm

- Sh compressor inlet 5 K
- Sh injection 5 K
- Subcooling 0 K
Results and discussion
Results and discussion

Intermediate Conditions ($\text{Sh}_{\text{inj}}=5$ K)

Injection mass flow rate (g/s) vs. Intermediate Pressure (bar)

- $T_c=40^\circ\text{C}$ (minj)
- $T_c=50^\circ\text{C}$ (minj)
- $T_c=60^\circ\text{C}$ (minj)
- $T_c=67^\circ\text{C}$ (minj)
- $T_c=40^\circ\text{C}$ (Pint)
- $T_c=50^\circ\text{C}$ (Pint)
- $T_c=60^\circ\text{C}$ (Pint)
- $T_c=67^\circ\text{C}$ (Pint)

Te ($^\circ\text{C}$) vs. Injection mass flow rate (g/s)

- $T_c=40^\circ\text{C}$ (minj)
- $T_c=50^\circ\text{C}$ (minj)
- $T_c=60^\circ\text{C}$ (minj)
- $T_c=67^\circ\text{C}$ (minj)
Results and discussion

Correlation between the intermediate conditions

Linear correlation between \( \frac{\dot{m}_{\text{inj}}}{\dot{m}_{\text{evap}}} \) and \( \frac{P_{\text{int}}}{P_{\text{evap}}} \) (small influence of injection sh for small sh)

\[
\frac{\dot{m}_{\text{inj}}}{\dot{m}_{\text{evap}}} = A + B \frac{P_{\text{inj}}}{P_{\text{evap}}}
\]

A = -0.366; B = 0.322

This correlation permits to close the system and estimate the intermediate conditions when the compressor works in different operating conditions.
Results and discussion

Evaluation of the correlation between the intermediate conditions

- A different compressor, but of the same model, was tested in a **heat pump with economizer** installed in a climatic chamber. The economizer size was the one indicated by the manufacturer in the installation guide.

- The intermediate conditions were fixed with a thermostatic valve of **5 K of superheat**.
Results and discussion

**EES model**

---Parameters of the cycle---

Refrigerant:

- \( R_407C \)

### Working conditions

- Condensation temperature: \( T_c = 50 \)°C
- Evaporation temperature: \( T_e = -17 \)°C
- Compressor inlet superheat: \( sh = 10 \)
- Injection superheat: \( SH_f = 5 \)
- Subcooling: \( sc = 5 \)

### Correlation

- Intermediate condition correlation:
  
  \[ m_i = A + B \cdot m_e \]

  where:
  
  \[ A = -0.365 \]
  
  \[ B = 0.322 \]

- Saturation temperatures:
  
  - Evaporation point (1):
    
    \( T_e = T(Refr, P = Pe, x = 1) \)
  
  - Condensation point (2):
    
    \( T_c = T(Refr, P = Pc, x = 1) \)
  
  - Injection point (3):
    
    \( T_i = T(Refr, P = Pm, x = 1) \)

### Thermal properties of different points

- Point (5) condenser outlet:
  
  \[ P_5 = P_e \]
  
  \[ T_5 = T_c - sc \]
  
  \[ h_5 = h(Refr, T = T_5, P = P_5) \]

- Point (6) economizer outlet-liquid side:
  
  \[ P_6 = P_e \]
  
  \[ T_6 = T_c \]
  
  \[ h_6 = h(Refr, T = T_6, P = P_6) \]

- Point (7) economizer outlet-vapor side:
  
  \[ P_7 = P_m \]
  
  \[ h_7 = h_5 \]

### Heat exchanger data

- Economizer:
  
  \[ \phi = 1.17 \]
  
  \[ \eta_w = 0.05997 \] [kg/s]
  
  \[ \eta_e = 0.04213 \] [kg/s]

- Properties:
  
  \[ \eta_v = 0.88 \] volumetric efficiency

- Mass flow rate:
  
  \[ m_i = 0.01384 \]
  
  \[ m_i = 0.1384 \]

- Inlet pressure:
  
  \[ P_f = P_m \]

- Inlet enthalpy:
  
  \[ h_f = h_f \]

- Heat exchanger data:
  
  \[ H = 0.39 \] plate height
  
  \[ W = 0.072209 \] plate width
  
  \[ d_p = 0.002082 \] plate separation
  
  \[ t_p = 0.0002318 \] plate thickness

- Number of plates:
  
  \[ N_p = 14 \]

- Area correction factor:
  
  \[ \phi = 1.17 \]

- Plate corrugated:
  
  \[ \beta = 120 \]

- Refrigerant:
  
  \( \text{R}407C \)

- Solution:

  \[ P_m = 523.40061 \] [kPa]

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Results and discussion

Comparison between measured data and predicted data

A quadratic function can fit better the results, but unlike the AHRI polynomials, with a linear function, it obtains a good fit for the intermediate conditions.
Conclusions
Conclusions

- The characterization methodology of vapor-injection compressors uses a modified calorimetric bench in order to add the injection line.
- The system is able to control the intermediate pressure and injection superheat independently.
- A scroll compressor with vapor-injection was tested over a wide range of evaporating and condensing pressures. For each condition a value of an intermediate pressure and intermediate superheat was fixed.
- For each test, compressor energy consumption, evaporator mass flow rate, and injection mass flow rate was recorded.
- The injection mass flow rate was correlated with the injection pressure.
- The correlation was used to predict the intermediate conditions in which an injection compressor installed in a heat pump with economizer will work.
- The obtained characterization seems to be only dependent on the used compressor.
Thanks for your attention