Modeling of Thermodynamic Processes for One Stage Refrigeration Systems with Scroll Compressors

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Abstract
The model which is presented in this paper was sustained by two numerical models: scroll compressor and air cooled evaporator/condenser equipment. By using this model, it is possible to obtain the thermodynamic process’s energy efficiency performances for the analyzed system, in a simple way and with less input parameters. With the help of this model it was possible to have polynomial equations and graphics in order to help us find the performances of the simulated refrigeration system. With these polynomial equations it is possible to verify the system in different working conditions. This model can be generalized to all one stage refrigeration systems, to test and verify the energy efficiency performances and the problems that may occur during the working period.

Keywords – compressor, evaporation, eco-efficient system, mechanical power, pressure ration

1. Introduction

Both the major energetic crisis and the global warming, which influence the worldwide economy and the future of the society, determine the development of energetic and ecological performances of both the refrigeration equipment and air conditioning systems. With this purpose in view, there is a worldwide supported effort made in order to decrease the carbon dioxide emissions resulted from the burning of fossil fuels and the other greenhouse effect gas emissions.

On this line, the usage of vapor-injected scroll compressors, digital scroll compressors and ecological refrigerants, leads the way too many applications in the energetic and mechanical engineering fields.

Those compressors are characterized by sustainability, efficiency, low noise and vibration level due to the short number of moving spare parts, mainly in relative friction.

The “Contributions to the study regarding the increase of eco-efficiency in refrigeration systems” Thesis is remarkable due to the fact that it is based on an applied research with immediate results related to the refrigeration systems.

2. The mathematical model
Mathematical simulation of the thermodynamic processes associated with the digital scroll compressor refrigeration system is thoroughly presented in the present Thesis.

The initial parameters are: condensation and evaporation pressure, overheating and sub-cooling temperature, room temperature, air temperature on both the condenser inlet/outlet line and evaporator inlet and outlet, as well as condenser, compressor and evaporator specific parameters.

Parameters resulted according to mathematical calculations are: compressor wall temperature, refrigerant temperature at the compressor discharge line, refrigerant weight rate in the refrigeration plant, compressors mechanical power, compressor volumetric efficiency, compressor isentropic efficiency, compressor compression ratio, heat conveyed by the compressor to the environment, refrigerant loss in the compressor, overall heat-transfer coefficients for each component of the refrigeration plant (compressor, condenser and evaporator, COP, condenser and evaporator efficacy, number of heat-transfer units for both the evaporators and the condenser, evaporation and condensation refrigerating capacity and refrigerant’s temperature on the evaporator inlet.

Mathematical simulation model of the refrigeration system with digital scroll compressor allows determining the thermodynamic efficacy and the properties in a simple manner and with a few parameters.

The numerical model symbols and process according to diagram from Figure 1 are:
- i-i₁=theoretical pressure drop to improve the refrigerant mass flow prediction at compressor suction point.
- i₁-i₂=refrigerant superheating at compressor suction point using a fictive semi-isothermal wall.
- i₂-e₂=mixing the refrigerant with compressor loses.
- i₂-in=irreversible adiabatic compression.
- in-e₂=adiabatic compression at constant volume.
- e₂-e₁=refrigerant de-superheating at compressor discharge point using a fictive semi-isothermal wall.
- e₁-e= theoretical pressure drop to improve the refrigerant mass flow prediction at compressor discharge point.
The main equations which describe the model are:

Refrigerant mass flow in i point given by:
\[ \dot{Q}_{th;cp} = \dot{Q}_{af} + \dot{Q}_{leak;cp} \]  
(1)

Where: \( \dot{Q}_{th;cp} \) – refrigerant mass flow in throttle process [kg/s]; \( \dot{Q}_{af} \) – refrigerant mass flow [kg/s]; \( \dot{Q}_{leak;cp} \) – refrigerant mass flow leakage [kg/s].

Pressure ratio for isentropic process is:
\[ \pi_{in;cp} = \frac{p_{in;cp}}{p_{i2;cp}} \]  
(2)

Where: \( p_{in;cp} \) – compressor inlet pressure [bar]; \( p_{i2;cp} \) – compressor pressure at i2 point [bar];

Refrigerant mass flow leakage trough compressor parts are given by:
\[ \dot{Q}_{leak;cp} = \frac{V_{leak;cp}}{v_{thr;cp}} \]  
(3)

Where: \( v_{thr;cp} \) – compressor throttle specific volume [kg/m³]; \( V_{leak;cp} \) – compressor volume of leakage [m³/s];

General equilibrium equation is given by:
\[ 0 = \dot{W}_{loss;cp} - \Phi_{i1;cp} + \Phi_{e2;cp} + \Phi_{amb;cp} \]  
(4)

Where: \( \dot{W}_{loss;cp} \) – electrical losses [W]; \( \Phi \) – cooling/heating power [W]; \( \Phi_{amb;cp} \), \( \Phi_{i1;cp} \), \( \Phi_{e2;cp} \) – thermal power of compressor in different points [W]

The compressor efficiency ratio is given by:
\[ \eta_{cp} = \frac{h_{e;cp} - h_{i;cp}}{W_{cp}} \]  

(5)

Where: \( W_{cp} \) – mechanical power [W]; \( h \) – enthalpy in different points [kJ/kg]; \( \eta \) - compressor efficiency [-];

Compressor pressure ratio is given by:

\[ \pi_{cp} = \frac{P_{e;cp}}{P_{i;cp}} \]  

(8)

Condenser capacity is given by:

\[ \Phi_{cd} = \dot{Q}_{e;cp} \cdot (h_{e;cp} - h_{e;cd}) \]  

(9)

The condenser efficiency ratio is given by:

\[ \eta_{cd} = 1 - e^{(-NTU_{cd})} \]  

(10)

Evaporator capacity is given by:

\[ \Phi_{vap} = \dot{Q}_{i;cp} \cdot (h_{i;cp} - h_{i;vap}) \]  

(11)

The evaporator efficiency ratio is given by:

\[ \eta_{vap} = 1 - e^{(-NTU_{vap})} \]  

(12)

The performance coefficient is given by:

\[ COP = \frac{\Phi_{vap}}{W_{cp}} \]  

(13)

The parameters specific to each equipment from the refrigeration system can be observed in Table 1.

<table>
<thead>
<tr>
<th>( V_{\text{exp}} )</th>
<th>( \text{AU}_{\text{exp},n} )</th>
<th>( \text{AU}_{\text{exp},n} )</th>
<th>( \text{AU}_{\text{amb},\text{exp},n} )</th>
<th>( \dot{Q}_{\text{af},\text{exp},n} )</th>
<th>( r_{\text{v,in},\text{exp}} )</th>
<th>( T_{\text{pierder},\text{exp}} )</th>
<th>( \alpha_{\text{exp}} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \text{m}^3/\text{h} )</td>
<td>( \text{W K}^{-1} )</td>
<td>( \text{W K}^{-1} )</td>
<td>( \text{W K}^{-1} )</td>
<td>( \text{kg s}^{-1} )</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>23,4</td>
<td>25</td>
<td>16</td>
<td>4,7</td>
<td>0,091</td>
<td>4,16</td>
<td>0,045</td>
<td>0,24</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>( \text{d}_i )</th>
<th>( \text{d}_e )</th>
<th>( \text{N} )</th>
<th>( m_{\text{cd}} )</th>
<th>( m_{\text{vap}} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \text{mm} )</td>
<td>( \text{mm} )</td>
<td>( \text{1/min} )</td>
<td>( \text{m}^3/\text{s} )</td>
<td>( \text{m}^3/\text{s} )</td>
</tr>
<tr>
<td>9,5</td>
<td>4</td>
<td>2900</td>
<td>1,8</td>
<td>1,13</td>
</tr>
</tbody>
</table>

Where: \( m_{\text{cd}} \) – air flow rate for the condenser (m\(^3\)/s); \( m_{\text{vap}} \) – air weight rate for the evaporator (m\(^3\)/s)
3. The validation process

The results achieved at the end of the simulation of the refrigeration system are rendered graphically below. These results consist in polynomial equations specific to the refrigeration system with digital scroll compressor. These results are based on the specific parameters of the analyzed refrigeration plant. Different calculation polynomials can be determined by using this simplified model. These polynomials are available for the refrigeration system associated with the experimental stand from the Compound of Laboratories in Colentina. They can be used on other systems too, in case of given coefficients.

$$\eta = \frac{A}{N_{rot}} + B \cdot \ln(N_{rot}) + C \cdot \ln(N_{rot})^m + D \cdot \frac{N_{rot}}{\ln(N_{rot})^n} + E \cdot e^{N_{rot}} + F \cdot \ln(N_{rot})^p$$

(14)

$$\dot{W}_{cp} = \frac{A}{N_{rot}} + B \cdot \ln(N_{rot}) + C \cdot \ln(N_{rot})^m + D \cdot \frac{N_{rot}}{\ln(N_{rot})^n} + E \cdot e^{N_{rot}}$$

(15)
Where A, B, C, D, E and F are coefficients [-]

4. Validation of mathematical model for digital scroll compressor

According to Figure 5, 6 and 7, the maximum relative errors are: 2% for refrigerant mass flow, 1.70% for compressor mechanical power and 1.60% for compressor discharge temperature.

Fig.5 Theoretical refrigerant mass flow (Qaf) vs. Experimental refrigerant mass flow (Qaf,m)

Fig.6 Theoretical mechanical power (Wcp) vs. Experimental mechanical power (Wcp;m)
5. Validation of mathematical model for fins and tubes evaporator

In the next figures (Figure 8 and Figure 9) the errors which result from the mathematical model values and the experimental ones are shown. The errors are represented by graphical view for evaporator refrigerant mass flow and capacity.

In Figure 8 we can see the errors between theoretical refrigerant mass flow and the experimental one. The error is around 4%.

In Figure 9 we can see the errors between theoretical and experimental evaporator capacity. The error is around 3%.
6. Conclusions

The refrigeration systems simulation models can forecast, with a maximum of 4% drift, the performances of the two single-stage vapor-compression systems.

The simulation models related to the scroll compressors can forecast, with a relatively ±1% drift, the refrigerant mass flow rate, the compressor discharge temperature and the mechanical work.

Mathematical simulation model can be used on any single-stage compression refrigeration system if the parameters of the system have been determined.

By using this simplified model, there could be determined different interpolating polynomials for the calculation of the refrigeration system performances.

According to the results achieved for the simulation of the two types of evaporator resulted the following:

- The mechanical power of the compressor increases along with the engine speed;
- Condenser thermal power on the air side increases along with the fan speed used for air circulation through condenser;
- The overall heat-transfer coefficient and condenser efficacy in the refrigeration system are ascending along with the number of heat-transfer units.
- The validation of the mathematical simulation model was fulfilled properly based on the data resulted from the experiments.
- For each type of analyzed equipment were set interpolating polynomials which allow calculating some parameters and coefficients related to the flow models in scroll compressors, evaporators and condensers.
- The determination of interpolating polynomials equations for the equipment associated with the analyzed experimental model
- Extension of the mathematical simulation model for different types of scroll compressors by improving the mathematical model developed;
• The development of calculation diagrams related to the components of a refrigeration system in order to determine the parameters in different operating conditions of the refrigeration plants used in the simulation;

References