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ENERGY AND EXERGY ANALYSES OF THE SCROLL COMPRESSOR

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ABSTRACT

This paper describes a computer code for analyzing scroll compressor loss mechanisms. The code includes both an energy analysis and an entropy (exergy) analysis. The analyses account for losses in the motor, friction, mixing, heat transfer, and the compression process. An experimental investigation was conducted to obtain data for model validation. Analytical and experimental results are presented and compared for several operating conditions and compressor flow configurations.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \dot{W}_{in} )</td>
<td>compressor input power</td>
</tr>
<tr>
<td>( e_x )</td>
<td>exergy per unit mass</td>
</tr>
<tr>
<td>( h )</td>
<td>enthalpy per unit mass</td>
</tr>
<tr>
<td>( i )</td>
<td>irreversibility per unit mass</td>
</tr>
<tr>
<td>( m )</td>
<td>mass flow</td>
</tr>
<tr>
<td>( Q )</td>
<td>heat transfer rate</td>
</tr>
<tr>
<td>( s )</td>
<td>entropy per unit mass</td>
</tr>
<tr>
<td>( T )</td>
<td>absolute temperature</td>
</tr>
<tr>
<td>( v )</td>
<td>velocity</td>
</tr>
<tr>
<td>( w )</td>
<td>work per unit mass</td>
</tr>
<tr>
<td>( \dot{W}_s )</td>
<td>shaft power</td>
</tr>
<tr>
<td>( \delta )</td>
<td>flow split parameter</td>
</tr>
</tbody>
</table>

Subscripts

- \( b \) bearing
- \( \text{dome} \) discharge side of the shell
- \( j \) inward stream
- \( k \) outward stream
- \( p \) pump (scroll elements)
- \( \text{out} \) total compressor shell
- \( \text{pri} \) primary suction path
- \( \text{sec} \) secondary suction path
- \( \text{sep} \) separator plate
- \( \text{shell} \) suction side of shell
- \( \text{tot} \) total
- \( o \) reference state
- \( 1-7 \) identified state points

INTRODUCTION

In order to maximize compressor efficiency, it is necessary to identify the processes which contribute to the overall loss. As compressor efficiencies have increased, identification of compressor losses has become more difficult. Loss minimization is further complicated when the impact of potential efficiency improvements on other characteristics such as noise, reliability and manufacturability is considered. Loss analysis is typically performed by evaluating the energy or EER (or COP) decrement for each loss mechanism. Losses may also be evaluated on the basis of entropy generation.
The objective of this research was to develop and validate an analytical model for performing both energy and entropy analyses of a low-side scroll compressor. The exergy method described in the next section was used to conduct the entropy analysis. Following a brief explanation of the exergy method, the analytical model is described and the results are compared with experimental data.

**EXERGY ANALYSIS**

The method of exergy analysis is a means of simultaneously applying the first and second laws of thermodynamics to engineering systems. Exergy is defined as the work available in a substance as a result of the nonequilibrium of the substance relative to a reference state. Exergy is synonymous with availability. While a first law analysis provides information about the quantity of energy consumed by losses, the exergy method affords the opportunity to rate the quality of energy lost. Only a brief definition of the exergy method will be presented here. An explanation of the method may be found in References [1-6]. McGovern and Harte [7] have applied the exergy method to the analysis of reciprocating compressors.

The flow exergy is a thermodynamic property which is a measure of the maximum work that can be extracted from a flow if it is brought to equilibrium with the reference state. Flow exergy is defined as

\[
ex = h - h_o - T_o(s - s_o) \tag{1}\]

where the subscript \(o\) denotes the reference state. For this investigation the reference state was taken to be standard atmospheric pressure and temperature. Exergy, like energy, can be transferred across a control volume by mass flow, heat transfer, and work. However, exergy differs from energy in that it is destroyed in irreversible processes. An exergy balance for a control volume assuming steady flow is

\[
\text{exergy destroyed} = \text{exergy in} - \text{exergy out} \tag{2}
\]

or

\[
i = \text{ex}_j - \text{ex}_k + \int \frac{(T-T_o)}{T}dQ/\dot{m} - w + v_j^2/2 - v_k^2/2 \tag{3}\]

The term \(i\) represents the quantity of exergy destroyed in irreversible processes. The integral term represents the exergy transfer associated with heat transfer across the surface of the control volume. Equation (3) may be used to identify and quantify loss mechanisms once the energy and mass conservation equations have been solved to give the energy fluxes and the thermodynamic state at each point in the system.

**ANALYTICAL MODEL**

The analytical model used for this investigation is an extension of the thermal model of the scroll compressor first outlined in Reference [8]. The model, shown schematically in Figure 1, includes the effects of motor losses, mechanical friction (bearings), and heat transfer. The suction flow is divided into two streams inside the compressor. One of the streams, the "secondary" suction flow, is used to cool the motor, oil, and bearings. This secondary
flow is then mixed with the "primary" suction flow prior to entering the scroll compression process. The flow split parameter $\delta$ indicates the ratio of primary flow to total mass flow

$$\delta = \frac{\dot{m}_{\text{pri}}}{\dot{m}_{\text{tot}}}$$

(4)

The global steady-state energy balance for the compressor is

$$\frac{\dot{E}_{\text{in}} - \dot{Q}_{\text{tot}}}{\dot{m}} = h_7 - h_1$$

(5)

where $\dot{Q}_{\text{tot}}$ is the total heat loss to ambient ($\dot{Q}_{\text{shell}} + \dot{Q}_{\text{dome}}$). The change in enthalpy may be expressed as the sum of the changes in enthalpy during the suction process (1-2), the compression process (2-5), and the discharge process (5-7):

$$\Delta h_{7-1} = \Delta h_{7-5} + \Delta h_{5-2} + \Delta h_{2-1}$$

(6)

$$\Delta h_{2-1} = (\dot{Q}_{\text{pri}} + \dot{Q}_{\text{sec}})/\dot{m}_{\text{tot}}$$

(7)

$$= (\dot{m}_{\text{pri}} \Delta h_{2\text{pri-1}} + \dot{m}_{\text{sec}} \Delta h_{2\text{sec-1}})/\dot{m}_{\text{tot}}$$

$$\Delta h_{5-2} = (\dot{W}_s - \dot{Q})/\dot{m}_{\text{tot}}$$

(8)

$$\Delta h_{7-5} = - (\dot{Q}_{\text{dome}} + \dot{Q}_{\text{sep}})/\dot{m}_{\text{tot}}$$

(9)

The mass flow rate and power input were obtained from experimental measurements. The mechanical and electrical losses were modeled using efficiencies which were obtained from experimental data. The compression process was modeled as a polytropic process. Values of the polytropic exponents and heat transfer coefficients were assumed from experimental data.

Energy balances were developed for the vapor in the suction side of the compressor, the suction portion of the shell, the compression process, and the discharge plenum. All calculations were performed using real gas properties for a typical refrigerant. Since the specific heat of the vapor may vary by 25 percent between suction and discharge, the set of equations required an iterative solution for enthalpy and shell temperature.

**EXPERIMENTAL APPARATUS**

A Carrier Millenium scroll compressor in a bolted shell was the test unit for this investigation. An instrumentation ring was used to facilitate routing the instrumentation out of the compressor. The test unit differed from a welded, hermetic shell in that the bolted shell had a larger surface area, a larger volume, and reduced heat conduction paths. These differences are not expected to significantly alter the characteristics of the compressor. The suction flow plumbing was modified to allow external control of the flow split between the secondary and primary suction paths, as shown in Figure 2. The compressor was extensively instrumented with heat flux gauges and thermocouples. The heat flux gauges were installed at several locations on the external shell. The thermocouples were installed to measure gas and metal temperatures throughout the compressor.

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The compressor was tested on a desuperheater test stand equipped with suction and discharge pressure transducers and thermocouples to determine operating conditions. The total mass flow was calculated from pressure and temperature measurements using a venturi. A mass flow meter was used to measure the secondary suction flowrate. All tests were conducted using a typical refrigerant. The compressor was tested at four operating conditions; three values of $\delta$ were tested at each condition.

RESULTS

Figures 3 and 4 show the trends of the experimental and analytical results for the temperature at the exit of the secondary suction flow ($T_{\text{sec}}$) and at discharge ($T_7$), respectively. Results are presented for four compressor operating conditions: 45/100/65 (saturated evaporating temperature/saturated condensing temperature/suction temperature), 45/130/65, 55/110/75, and 0/90/20. All temperature values are in degrees F. The results indicate that the temperature at the exit of the secondary suction flow increases with increasing $\delta$, while the discharge temperature remains nearly constant for the range of $\delta$ tested. Detailed analysis showed agreement between analytical and experimental results to within 4 degrees F for $T_{\text{sec}}$. The agreement for $T_7$ was within 5 degrees F for 45/100/65, 45/130/65, and 55/110/75. The calculated and experimental results for $T_{\text{sec}}$ did not agree well for 0/90/20. This discrepancy is thought to be due to an inadequate model of the backflow which occurs during the discharge process when operating at some off-design conditions.

A comparison of the energy and exergy losses for several processes included in this model is shown in Figure 5 for a typical operating condition. The energy and exergy losses for the motor and friction are equal because all of the inefficiencies from these two processes were assumed to be unavailable for useful work. The mixing process was assumed to be constant pressure, therefore the energy loss during mixing was zero. The loss for the compression process as calculated using the energy method is 25 percent higher than that calculated using the exergy method. The fact that the energy compression loss is greater than the exergy loss is a result of definition. The energy loss is defined as the difference between the energy which was used in the actual compression process and the energy which would have been used if the process were isentropic. The exergy loss is the irreversibility in the actual compression process as calculated using Equation (3). Both the energy and exergy analyses indicate that the largest loss is due to the inefficiency of the motor and that friction losses and compression losses are of the same magnitude.

CONCLUSION

A computer model for analyzing the energy and exergy losses in scroll compressors was developed. Calculated and experimental results for the thermal processes were compared and were in agreement for most operating conditions. The model was used to calculate the energy and exergy losses for several processes within the compressor. This analytical tool can be used to optimize compressor designs.
REFERENCES


Figure 1 Analytical Model

Figure 2 Schematic of Test Compressor
Figure 3 Comparison of Measured and Calculated Secondary Suction Exit Temperature ($T_{2\text{sec}}$) vs. Flow Split Parameter ($\delta$)

Figure 4 Comparison of Measured and Calculated Discharge Temperature ($T_7$) vs. Flow Split Parameter ($\delta$)

Figure 5 Comparison of Typical Energy and Exergy Losses for Several Processes