TURBOMACHINERY & PUMP SYMPOSIA | VIRTUAL
DECEMBER 8-10, 2020
SHORT COURSES: DECEMBER 7, 2020

A NOVEL METHOD FOR EVALUATING SIDELOAD COMPRESSOR THERMODYNAMIC PERFORMANCE

Mark R. Sandberg, P.E.
Principal Consulting Engineer
Sandberg Turbomachinery Consulting, LLC
Montgomery, Texas, USA

Mark R. Sandberg is the principal of Sandberg Turbomachinery Consulting, LLC. Before forming this consulting practice in 2016, he was a Consulting Machinery Engineer with Chevron Energy Technology Company in Houston, Texas for more than fifteen years. Prior to joining Chevron, he was employed by ARCO, Petro-Marine Engineering, and The Dow Chemical Company. During his sixteen years at ARCO, he was involved with a number of gas turbine driven compressors, both internationally and on the North Slope of Alaska. Throughout the majority of his career, he has been involved in providing technical assistance and services associated with the selection, design, specification, procurement, and testing of new rotating equipment along with failure analysis and troubleshooting issues with existing equipment, primarily in upstream oil and gas production and LNG processing operations. Mark has more than 40 years of varied experience in the process industries and has been involved with the design, manufacture, testing, and installation of several small to large gas turbine driven centrifugal compressor trains worldwide. Mr. Sandberg has B.S.M.E. and M.S. (Mechanical Engineering) degrees from the University of Illinois at Urbana-Champaign, is a registered Professional Engineer in the State of Texas, an Emeritus Member of the Texas A&M Turbomachinery Advisory Committee, a member of AIAA and a Life Fellow Member of ASME.

ABSTRACT

A method has been developed that allows reasonably accurate sectional performance evaluation of sideload compressors based upon only external measurements available from the streams entering and exiting the compressor casing at the respective nozzles. Estimated sectional thermodynamic performance has historically been limited to acceptance testing where temporary temperature measurement instrumentation has been inserted into the flow path between sections. Accurate longer term evaluation and monitoring of compressor performance once installed and operating has not been possible. While overall, multi-section thermodynamic performance may be obtained by the heat balance method as defined in ASME PTC 10, individual section behavior has not been attainable with the existing procedures. The method logic and equation development is presented for a two-section sideload compressor but can be extended to compressors with additional sections (typically three or four) through the application of some additional logic and calculations. A number of example cases derived from data sheet predicted information and a PTC 10 Type 2 test are included to demonstrate the potential accuracy of the proposed method.

INTRODUCTION

Sideload compressors represent a unique design among the different configurations available for centrifugal compressors. They are well suited for refrigeration compression applications where successively available lower pressure levels exist at proportionately lower temperatures. This allows sidestream flows to be introduced to the mainstream flow within the casing of a multi-section sideload compressor without having to extract and cool the mainstream flow, effectively eliminating the pressure losses attributed to any external heat exchange equipment. A significant number of current and future applications of sideload compressors include two, three or four sections contained within a single compressor casing. Although any number of individual sections are theoretically possible, the most common configurations contain from two to four sections, usually with one or two stages per section. Process requirements will generally dictate the required number of sections for an application, but rotordynamic considerations may also limit the number of sections within a single casing due to bearing span limitations. The photograph in Figure 1 is an example of a typical two-section sideload compressor.

Figure 1. Typical Two-Section Sideload Compressor

The flow configuration of sideload compressors creates challenges associated with the initial and continued validation of predicted thermodynamic performance of these machines. A schematic of a four section sideload compressor is provided in Figure 2. Sidestream flows are introduced and mixed with the main stream flow within the compressor casing. This is contrary to other multi-section sidestream compressor designs where the sidestreams are mixed externally to the compressor casings. While both designs allow an overall energy balance to be completed, the sideload design does not allow determination of section-by-section energy balances due to the internal mixing of the streams. Sectional performance evaluation requires knowledge of the inlet and discharge pressures and temperatures along with gas composition and mass flow rate to calculate performance parameters such as head (work), efficiency and gas power. The preceding section discharge temperature or succeeding section mixed inlet temperature are not usually measurable with sideload compressor designs. Absence of this information does not allow sectional performance to be determined without some simplifying assumptions being made which may compromise the accuracy of any results. An exception to this limitation of sideload compressors exists in the case where the sidestream flow is an extraction instead of a mass flow addition to the mainstream flow of the machine. In this case, the preceding section and succeeding section temperatures are the same and equal to the extraction sidestream measured temperature. This temperature along with the sidestream pressure can allow the evaluation of the individual section performance parameters.

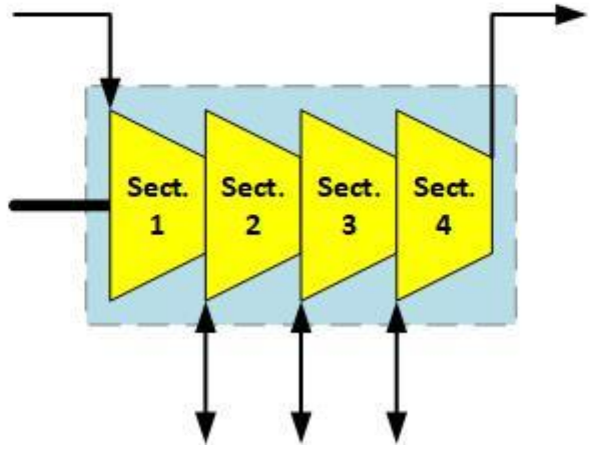

Figure 2. Four Section Sideload Compressor Schematic

A need to accurately determine sectional performance of a sideload compressor exists in order to demonstrate and verify predicted performance. ASME PTC 10, "Performance Test Code on Compressors and Exhausters," (1997) defines the requirement to evaluate this sectional performance. It states in paragraph 3.5.1, "Compressors having flows added or removed at intermediate locations between the inlet and final discharge are handled by treating the compressor by sections." A further addition and requirement is provided in paragraph 3.5.3 which states, "When the sidestream flow is inward, the discharge temperature of the preceding section shall be measured prior to the mixing of the two streams……..The discharge temperature is needed to compute the performance of the preceding section and to compute the reference mixed temperature for the next section inlet." This generally defines the need to add some temporary form of temperature measurement as illustrated in Figure 3 to allow accurate test results to be recorded in order to validate predicted performance of the individual sections. The temperature measurement must be placed between locations "A" and "B" in the attached diagram.

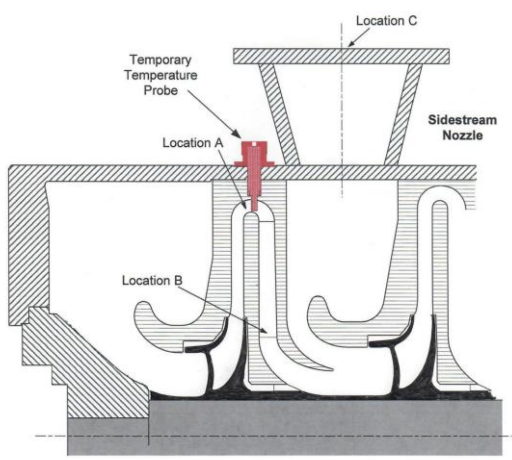

Figure 3. Two Section Sideload Cross-Sectional Test Diagram

Although the temperature measurement described above supplies the needed parameter to determine the sectional performance of the compressor, it is subject to significant uncertainties that need to be identified. The flow field that is present in the return bend and channel of the stationary components is complex and highly variable with variations in flow rate from surge to overload resulting in a temperature measurement that is subject to substantial error. Subsequent mixture and downstream temperature measurements are also impacted and alter calculated efficiencies which are relatively sensitive to measured total temperature. In the absence of any better measurement or method, the existing uncertainty in calculated performance of sideload compressors has been accepted as the most accurate.

Once the machines have completed any factory acceptance testing, the temporary test temperature instrumentation has been removed due to longer term reliability and maintenance considerations. The resulting lack of a temperature measurement leaves the end user without any reasonable way to estimate or monitor sideload compressor sectional performance. Predicted or "as tested" performance characteristics can be factored into continuous monitoring and operating analyses, but the sensitivity of compressor operating parameters to process variations can result in erroneous performance variation conclusions.

SOLUTION METHODOLOGY

A new evaluation method has been developed that provides reasonably accurate values of sideload compressor sectional performance. This method only requires values of pressures, temperatures and flow rates at each of the main casing nozzles, which are normally readily available external measurements. There is no need for an internal temperature measurement. This method can be applied to any number of sections within a single casing, however, the governing equations and method will only be presented for a two-section machine. Results from two, three, and four-section compressor analyses will be presented to demonstrate the relative accuracy of the method.

Sandberg and Colby (2013) introduced an alternate method for calculating compressor performance compared to the procedure developed by Schultz (1962) which is the basis of ASME PTC 10 (1997). The Sandberg-Colby method utilizes only values of enthalpy, entropy and temperature at section inlet and discharge conditions. Reported accuracies of results are comparable or better than the Schultz method at near ideal conditions (such as PTC 10 Type 2 testing) and typically superior as conditions move into the dense phase region. Sideload compressor operating conditions will normally fall within the near ideal conditions and at pressures below the cricondenbar and temperatures near or above saturated vapor temperatures.

The polytropic head (work) is given from the following relation:

$$w_p = \left(h_d - h_s\right) - \left(\frac{T_s + T_d}{2}\right)\left(s_d - s_s\right) \tag{1}$$

and the associated polytropic efficiency is:

$$\eta_p = \frac{w_p}{\left(h_d - h_s\right)} = 1 - \left(\frac{T_s + T_d}{2}\right)\left(\frac{s_d - s_s}{h_d - h_s}\right) \tag{2}$$

These relations will be utilized in the evaluation of sectional performance parameters in the following method development. The Schultz method could also be utilized, but is not as straightforward due to the necessary derivation of temperatures and pressures from the values of enthalpy and entropy that are directly available from the analyses.

An important behavior of a polytropic compression path (path of constant polytropic efficiency) and a major influence on the accuracy of the Sandberg-Colby method is illustrated in Figure 4 based upon a high stage section of a propane refrigeration compressor. The relative change in entropy versus enthalpy from inlet to discharge is estimated by the solid black line for both the Schultz and Sandberg-Colby endpoint methods. Also included are two numerical integrations of the Sandberg-Colby method where multiple steps of equal compression ratios and a constant polytropic efficiency are plotted for 10 steps and 50 steps. The nearly linear path of the multi-step methods confirms the assumption that the linear relation of the endpoint methods are accurate and that the simple average of inlet and discharge temperatures is also valid. A further examination of equation 2 demonstrates that the slope of the enthalpy-entropy diagram is inversely proportional to the value of the polytropic efficiency. In other words, a lower polytropic efficiency will result in a steeper entropy-enthalpy linear characteristic.

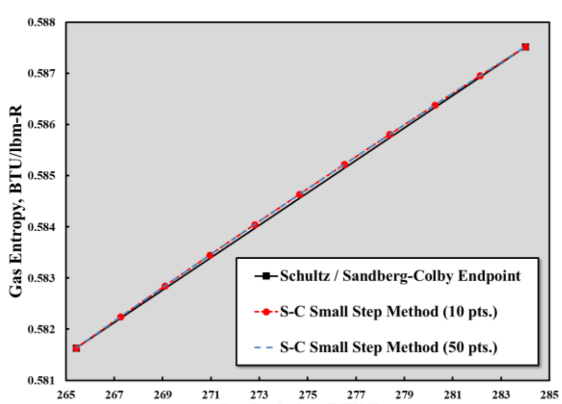

Figure 4. Enthalpy-Entropy Diagram for Propane Compression Stage

A comparison of different polytropic efficiencies and their impact on the enthalpy-entropy diagram is shown in Figure 5. It should be noted that the inlet conditions are constant and equal to that of Figure 4. In addition, the discharge pressures are also the same with only the discharge temperature being varied. The range of discharge temperature from the lowest to highest efficiency is only 10 F (5.56C). This indicates how sensitive entropy variations and efficiencies are to changes in temperature.

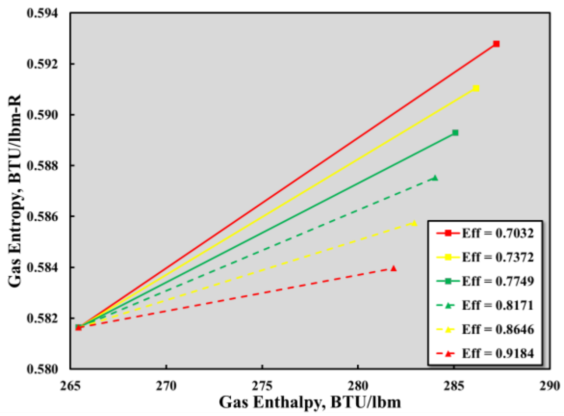

Figure 5. Comparison of Enthalpy vs. Entropy Changes for Various Efficiencies

The two-section sideload compressor method development will reference the schematic and variable identification that is provided in Figure 6. Although only a two-section analysis is developed in this document, additional logic and steps applied to the following analysis can result in the evaluation of three, four, or additional section sideload compressors.

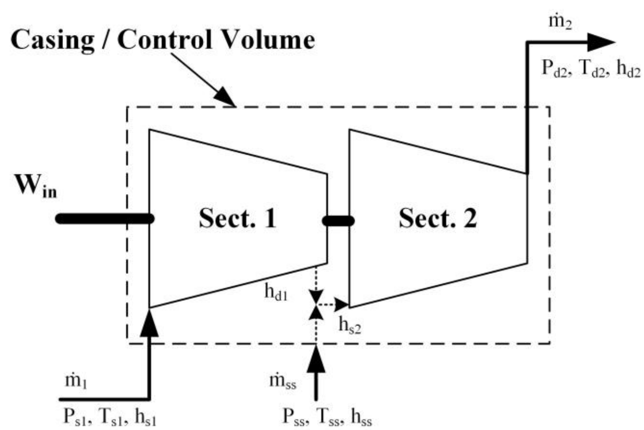

Figure 6. Two-Section Compressor Schematic

Mass Balance

Governing equation development starts with a mass balance of the sideload compressor inlet and sidestream flows.

$$\dot{m}_1 + \dot{m}_{ss} = \dot{m}_2 = \dot{m}_t \tag{3}$$

but, defining the mass balance in terms of mass fractions instead of mass flow rates:

$$\frac{\dot{m}_1}{\dot{m}_t} + \frac{\dot{m}_{ss}}{\dot{m}_t} = x_1 + x_2 = 1 \tag{4}$$

where,

$$x_1 = \frac{\dot{m}_1}{\dot{m}_t} \quad \text{and,} \quad x_2 = \frac{\dot{m}_{ss}}{\dot{m}_t} \tag{5}$$

Energy Balances
Energy (work) balances are an integral part of compressor performance calculations. ASME PTC 10 (1997) defines a number of different methods to determine consumed gas or shaft power of a compressor. As prescribed by PTC 10, this is normally done for a single section, but can be expanded to multi-section compressors if the additional sections are treated as additional parasitic losses. Sideload compressors create a unique situation for this multi-section case due to the fact that individual section inlet and outlet streams are not directly measurable. This precludes the separation of individual section performance without the addition of internal parameter measurements, such as the previously described preceding section discharge temperature.

Overall Balances
In the absence of internal temperature measurements, an overall energy balance of a sideload compressor will provide some insight into the total performance and any potential degradation of the machine. This analysis is based upon the application of the First Law of Thermodynamics to a control volume that is defined as the compressor casing as shown in Figure 6. Writing the balance in terms of consumed power:

$$PWR = \dot{m}_t \cdot W_{in} = (\dot{m}_1 + \dot{m}_{ss}) \cdot h_{d2} - \dot{m}_{ss} \cdot h_{ss} - \dot{m}_1 \cdot h_{s1} \tag{6}$$

Conversion of the power to the specific work input based upon the total mass flow rate:

$$W_{in} = h_{d2} - x_2 \cdot h_{ss} - x_1 \cdot h_{s1} \tag{7}$$

It is evident that this relation may be evaluated from externally measured parameters of pressure, temperature and mass flow at the various compressor nozzles. A similar equation can be developed for the overall lost work (entropy generation) from the Second Law of Thermodynamics that can be connected with the compressor.

$$\Delta S = s_{d2} - x_2 \cdot s_{ss} - x_1 \cdot s_{s1} \tag{8}$$

While these relations provide insight into changes in overall energy consumption and losses, it cannot provide information on variations in compressor performance due to changes in process conditions or sectional performance health.

Sectional Balances
Comparable enthalpy and entropy balances can be developed for the individual sections of the sideload compressor. Once again, starting with the total power consumption of the compressor and referencing Figure 6:

$$PWR = \dot{m}_t \cdot W_{in} = \dot{m}_1 \cdot (h_{d1} - h_{s1}) + (\dot{m}_1 + \dot{m}_{ss}) \cdot (h_{d2} - h_{s2}) \tag{9}$$

Converting the power to specific work input,

$$W_{in} = x_1 \cdot (h_{d1} - h_{s1}) + (x_1 + x_2) \cdot (h_{d2} - h_{s2}) = \sum_{i=1}^{n} \left[\left(\sum_{j=1}^{i} x_j \right) \cdot \left(h_{di} - h_{si} \right) \right] \tag{10}$$

A similar sectional balance can be written for the total and sectional entropy generation.

$$\Delta S = x_1 \cdot (s_{d1} - s_{s1}) + (x_1 + x_2) \cdot (s_{d2} - s_{s2}) = \sum_{i=1}^{n} \left[\left(\sum_{j=1}^{i} x_j \right) \cdot \left(s_{di} - s_{si} \right) \right] \tag{11}$$

Although these equations provide the PTC 10 required information on individual section performance, they cannot be directly solved because there are more unknown quantities than equations, namely h_{d1} and h_{s2} for the enthalpy balance and s_{d1} and s_{s2} for the entropy balance. The measurement of the first stage discharge temperature during factory acceptance testing along with the value of the sidestream nozzle inlet pressure allows the calculation of the enthalpy and entropy, h_{d1} and s_{d1}. Additional relations need to be developed that will allow the determination of the remaining unknown enthalpy and entropy terms.

Sidestream Mixing Relations
The sidestream mixing region provides a correlation between the second section suction conditions and first stage discharge properties. If conditions at either location are known, the other conditions can be derived with the addition of the known sidestream conditions. Reference to Figure 6 illustrates the following equation for the associated enthalpy values.

$$h_{s2} = \frac{x_1 \cdot h_{d1} + x_2 \cdot h_{ss}}{\left(x_1 + x_2\right)} \quad (12)$$

A similar relation can be developed for the specific entropy:

$$s_{s2} = \frac{x_1 \cdot s_{d1} + x_2 \cdot s_{ss}}{\left(x_1 + x_2\right)} \quad (13)$$

Once again, in the case of precisely measured first stage discharge total temperature, these equations may be solved to provide a complete set of performance parameters for each section of the sideload compressor, however, this is normally not the case for longer term operation of these machines. An alternative solution method can be developed to provide a reasonably accurate set of sectional performance parameters.

Equivalence of Overall and Sectional Balances

Overall and sectional enthalpy and entropy balances are repeated and expanded as follows:

$$W_{in} = h_{d2} - x_2 \cdot h_{ss} - x_1 \cdot h_{s1} = W_{in,1} + W_{in,2} \quad (14)$$

Equating this overall enthalpy balance to individual section balances,

$$W_{in,1} = y_1 \cdot W_{in} = x_1 \cdot (h_{d1} - h_{s1}) \text{ and } W_{in,2} = y_2 \cdot W_{in} = (x_1 + x_2) \cdot (h_{d2} - h_{s2}) \quad (15)$$

Where, it is obvious that,

$$y_1 + y_2 = 1 \quad (16)$$

Similarly, an overall balance and individual section balances can be written for entropy.

$$\Delta S = s_{d2} - x_2 \cdot s_{ss} - x_1 \cdot s_{s1} = \Delta S_1 + \Delta S_2 \quad (17)$$

with an initial assumption that,

$$\Delta S_1 = y_1 \cdot \Delta S = x_1 \cdot (s_{d1} - s_{s1}) \text{ and } \Delta S_2 = y_2 \cdot \Delta S = (x_1 + x_2) \cdot (s_{d2} - s_{s2}) \quad (18)$$

Unfortunately, no unique solution of this system of equations (overall and sectional balances along with mixing relationships) exists, rather there are an infinite number of solution values. Additionally, there is no verification that the variation in entropy follows the same factored relationship that has been defined for the enthalpy split between the individual sections. A solution requires establishment of some additional boundary conditions.

Sidestream Boundary Condition

An obvious constraint exists with the sidestream flow entering the compressor between sections. As stated previously, ASME PTC 10 (1997) prescribes that sideload compressors shall be tested as individual compressor sections. The pressure that exists at the sidestream nozzle provides another variable that can be utilized to determine thermodynamic performance for both the upstream and downstream sections. Functional relationships within thermodynamics allow the determination of a parameter as a function of two other independent thermodynamic parameters. The following relation can be developed with the sidestream pressure:

$$s = s(P, h) \quad (19)$$

More specifically, the following equation can be written for the first section of a two-section sideload compressor.

$$s_{d1a} = s\left(P_{ss}, h_{d1}\right) = s\left(P_{ss}, \left[h_{s1} + \frac{y_1 \cdot W_{in}}{x_1}\right]\right) \quad (20)$$

and, for the second section,

$$s_{s2a} = s\left(P_{ss}, h_{s2}\right) = s\left(P_{ss}, \left[h_{d2} - \frac{y_2 \cdot W_{in}}{\left(x_1 + x_2\right)}\right]\right) \tag{21}$$

These two additional relations allow unique solutions to be obtained for both sections of a two-section machine when the following parameters converge to the same values.

$$s_{d1} = s_{d1a} \quad \text{and} \quad s_{s2} = s_{s2a} \tag{22}$$

Values for y_1 and y_2 can be obtained by solving the resulting system of equations. Although the sum of these two parameters should equal unity, it can be demonstrated that this is not the case when the relations are solved separately. Of course, this sum can be forced to unity by setting the sum equal to unity and solving for the opposite parameter, but the result of this is that the sectional sum of entropy does not equal the overall entropy generation value. Conversely, if separately obtained values of y_1 and y_2 are used, the sectional sums of both enthalpy and entropy do not equate with overall balance values, but the relative errors of both are reduced in comparison with the entropy balance of the first method. Potential reasons for these imbalances will be discussed after some examples of the analyses are presented.

Setting Iteration Limits

The values of the fractional enthalpy factors, y_1 and y_2, must obviously fall between zero and unity. While an iterative solution of one or both of these quantities can numerically fall between these limits, a solution can be assured if reasonable extents are used. These limiting values can be established if practical limits of sectional compressor performance are estimated. Focusing on the first section, the lower limit of enthalpy increase in the section is represented by an isentropic solution to this first section. This is defined with the following values set for the first stage discharge conditions.

$$s_{d1} = s_{s1} \quad \text{and} \quad h_{d1,\min} = h(P_{ss}, s_{s1}) \tag{23}$$

Applying the resulting values of first section discharge parameters to sectional enthalpy and entropy deviations across the first section, a lower limit of the first section fractional enthalpy factor can be calculated.

$$y_{1,\min} = \frac{x_1 \cdot \left(h_{d1,\min} - h_{s1}\right)}{W_{in}} \tag{24}$$

Conversely, the maximum enthalpy deviation across the first section is represented by an isentropic compression path of the second section of the machine. Governing equations for this upper limit are provided by:

$$s_{s2} = s_{d2} \quad \text{and} \quad h_{s2,\max} = h\left(P_{ss}, s_{d2}\right) \tag{25}$$

Two additional equations are necessary to convert the above values to first section discharge conditions with relations for the mixing in the sidestream area.

$$h_{d1,\max} = \frac{\left(x_1 + x_2\right) \cdot h_{s2,\max} - x_2 \cdot h_{ss}}{x_1} \quad \text{and} \quad s_{d1,\max} = \frac{\left(x_1 + x_2\right) \cdot s_{s2,\max} - x_2 \cdot s_{ss}}{x_1} \tag{26}$$

with an upper limit of the first section fractional enthalpy factor calculated as:

$$y_{1,\max} = \frac{x_1 \cdot \left(h_{d1,\max} - h_{s1}\right)}{W_{in}} \tag{27}$$

Finally, the equivalent iteration limits can be defined for the second section as follows.

$$y_{2,\min} = 1 - y_{1,\max} \quad \text{and} \quad y_{2,\max} = 1 - y_{1,\min} \tag{28}$$

These relations allow solutions to be completed for both sections of a two-section sideload compressor. As noted previously, they are not necessarily exact but allow reasonably accurate and comparable or better solutions when evaluated against other methods. This will be demonstrated for a number of different examples. As mentioned previously, it is possible to expand this methodology to sideload compressors with a larger number of sections through the application of additional logic and calculation steps.

Solution Logic Summary

Two related but different solution methods are presented that allow determination of sectional performance of two-section sideload compressors. Each offers relatively accurate estimates of sectional performance when compared to baseline data. One method (individual iteration on enthalpy fractions) splits errors between enthalpy and entropy balances. The other (iteration on only first section enthalpy fraction) completely balances the enthalpy between sectional and overall balances but results in a larger difference between overall and sectional entropy balances along with the potential for the calculated second section suction pressure not being exactly equal to the sidestream pressure.

A step-by-step summary of each method is provided in the table below. It should be noted that a single column between the two methods designates common calculation procedures for both methods.

<table>
<tr><th>Step Number</th><th>Method 1:
$y_1 + y_2 \cong 1$</th><th>Method 2:
$y_1 + y_2 = 1$</th></tr>
<tr><td>1</td><td colspan="2">Complete overall enthalpy and entropy balances in accordance with equations 7 and 8.</td></tr>
<tr><td>2</td><td colspan="2">Calculate the minimum and maximum values of the enthalpy fractional splits with equations 23 through 28.</td></tr>
<tr><td>3</td><td colspan="2">Using a trial value of the first section enthalpy fraction, y_1, between y_{1min} and y_{1max}, calculate the following parameter values:
a) h_{d1} from equation 15
b) s_{d1} from equation 18
c) s_{d1a} from equation 20
d) entropy deviation: $\Delta s_{d1}=(s_{d1}-s_{d1a})$</td></tr>
<tr><td>4</td><td colspan="2">Iterate on the value of y_1 in step 3 until $\Delta s_{d1}=0$.</td></tr>
<tr><td>5</td><td colspan="2">Calculate remaining first section discharge conditions:
a) $P_{d1} = P_{ss} = P(h_{d1},s_{d1})$
b) $T_{d1}=T(h_{d1},s_{d1})$</td></tr>
<tr><td>6</td><td>Using a trial value of the second section enthalpy fraction, y_2, between y_{2min} and y_{2max}, calculate the following parameter values:
a) h_{s2} from equation 15
b) s_{s2} from equation 18
c) s_{s2a} from equation 21
d) entropy deviation: $\Delta s_{s2}=(s_{s2}-s_{s2a})$</td><td>Using a value of the second section enthalpy fraction, y_2, where $y_2 = (1 - y_1)$, calculate the following parameter values:
a) h_{s2} from equation 12 or 15
b) s_{s2} from equation 13 or 18
c) s_{s2a} from equation 21
d) entropy deviation: $\Delta s_{s2}=(s_{s2}-s_{s2a})$</td></tr>
<tr><td>7</td><td>Iterate on the value of y_2 in step 6 until $\Delta s_{s2}=0$.</td><td></td></tr>
<tr><td>8</td><td>Calculate remaining second section suction conditions:
a) $P_{s2} = P_{ss} = P(h_{s2},s_{s2})$
b) $T_{s2}=T(h_{s2},s_{s2})$</td><td>Calculate remaining second section suction conditions:
a) $P_{s2} = P(h_{s2},s_{s2})$
b) $T_{s2}=T(h_{s2},s_{s2})$</td></tr>
<tr><td>9</td><td colspan="2">Calculate remaining sectional performance parameters including:
a) Individual section polytropic head
b) Individual section polytropic efficiency
c) Overall and sectional gas power</td></tr>
</table>

Table 1. Solution Method Steps and Comparison

A representative example of a two section propane compressor is provided in the Appendix and is intended to act as a guide and confirmation of both methods presented above in Table 1. This hypothetical example includes all of the data needed to calculate the parameters utilized and derived from the methods and compare against first section discharge and second section suctions conditions along with recognized compressor performance parameters.

SOLUTION METHODOLOGY VALIDATION

Validation of the proposed new methodology to determine sideload compressor thermodynamic performance is critical to establishing its utility in accurately predicting compressor performance. This will be demonstrated through two different types of comparisons. First, an evaluation is presented that compares performance parameters derived from the method relative to information provided on equipment supplier predicted data sheets. Data sheet variables used to perform the analyses are limited to the identified external quantities that could be measured on an actual machine. Sectional enthalpy and entropy balances, along with calculated polytropic head and efficiency, are compared with the equivalent values derived from internal process parameters listed on the compressor data sheets.

Proposed method calculations presented are limited to the procedures listed as Method 1 in Table 1 above. This is due to the observed more favorable agreement between method and data sheet calculated values of sectional polytropic head and efficiency consistent with all of the example cases evaluated. A common equation of state to estimate thermodynamic properties is also considered important to minimize any differences in such properties. The NIST REFPROP, Lemmon, et al. (2018), software application was applied to both the method calculations presented herein and with an independent calculation of performance parameters provided from data sheets and test measurements. Finally, common performance calculation methods to derive polytropic head and efficiency as defined by Sandberg and Colby (2013) were applied to both data sheet and test conditions to eliminate any differences that could be due to any variations in calculation procedures.

Data Sheet Evaluations

Two different performance parameter categorizations were examined with respect to data sheets. One was a comparison between overall and sectional enthalpy and entropy balances for both listed data sheet parameters and those derived from the proposed method. The overall versus sectional balance differences are expressed as percentage deviations according to the following relation:

$$\%Deviation = \left(\frac{Parameter_{section} - Parameter_{overall}}{Parameter_{overall}} \right) \cdot 100 \tag{29}$$

A second, similar comparison was completed using individual section compressor performance parameters of polytropic head and polytropic efficiency of the proposed simulation method versus data sheet information. This evaluation was also completed on a percentage deviation basis and can be expressed generally as:

$$\%Deviation = \left(\frac{Parameter_{simulation} - Parameter_{datasheet}}{Parameter_{datasheet}} \right) \cdot 100 \tag{30}$$

These parameter assessments are considered to be the most influential in demonstrating the validity and accuracy of the proposed performance calculation method. The comparison of overall and sectional enthalpy and entropy balances provides significant insight into the accuracy of the sectional calculations since they should equate to the overall balances. Differences between the sectional and overall entropy balances indicate deviations in estimated losses, or inefficiencies, of the sectional calculations. Once again, the overall and sectional entropy balances should theoretically be equal. The values of enthalpy and entropy were derived from the inlet and discharge total pressures and total temperatures provided on the respective data sheets.

Polytropic head and efficiency are fundamental compressor performance parameters that allow comparison of compressor performance or changes in performance among and between compressor sections. ASME PTC 10 (1997) includes these parameters as important measures in the evaluation of thermodynamic performance. These parameters utilize values of enthalpy and entropy as is illustrated in equations 1 and 2 above.

Two-Section Example Cases

Seven distinct case examples are included in the two-section data sheet evaluation. All but one of these are propane refrigeration compressors with designs that incorporate different pressure and temperature levels and different sectional mass flow rate fractions. A comparison of overall and sectional enthalpy and entropy balances based upon data sheet process conditions and the solution method (simulation) based upon casing nozzle pressures, temperatures and flows is provided in Figure 7.

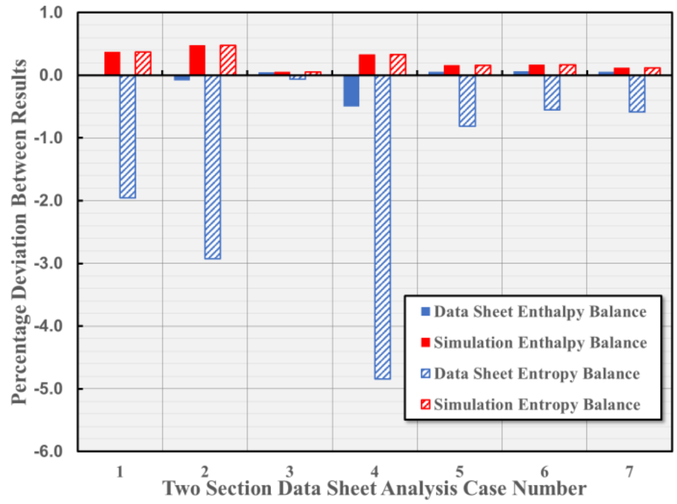

Figure 7. Two-Section Deviation Between Overall and Sectional Energy Balances

An examination of this figure shows that the overall and sectional enthalpy balances for the data sheet information displays minimal deviation. A slightly more notable deviation is evident for the simulation, but all deviations are less than 0.5 percent. It should be noted that it has previously been mentioned that the use of calculation method 1 results in a deviation in the enthalpy balance, whereas method 2 produces no deviation in enthalpy but a higher deviation in the entropy balance. More significant deviation is reflected in the data sheet overall versus sectional entropy deviation. This consistently negative deviation is indicative of lower losses with most sectional calculations, or likely overstated efficiencies when compared against the overall balances. Simulation entropy balances are comparable to the enthalpy balances and lower than positive 0.5 percent which suggests a slight underestimate of efficiency.

Figure 8. Two-Section Deviation Between Data Sheet and Simulation Polytropic Head and Efficiency

Figure 8 provides a comparison for sectional values of polytropic head and efficiency between the simulation method and data sheet derived properties. Very close agreement of polytropic head is evident. There is more pronounced deviation in the sectional estimates of the polytropic efficiency. The significant difference displayed in case 7 deserves some additional explanation. A review of the data sheet values uncovered some inconsistencies in the sectional process parameters and derived properties of head and efficiency. The independently calculated values using the common equation of state and data sheet pressures and temperatures also showed a potential issue with the data sheet. First section discharge and second section inlet temperatures showed a deviation of 5°F (2.5C) and 2°F (1C), respectively, between the simulation method and data sheet. This is greater than any of the other cases examined. Assuming that this example is an outlier, it is observed that differences in polytropic head is within 1 percent and polytropic efficiency is within 5 percent among the included cases. Another observation is that the simulation method more frequently overestimates the first section efficiency and underestimates the second section efficiency.

Evaluation of the two-section data sheet comparison demonstrates that the proposed alternate calculation method results in fairly accurate estimates of sideload compressor performance. Polytropic head is accurately predicted with the proposed method. While there is more deviation evident in the predicted values of polytropic efficiency, recognition of the sensitivity of the efficiency to slight variations in temperatures listed on data sheets and estimates of these temperatures from individual stage performance modeling and any possible inter-stage corrections could be possible causes of the increased differences.

Three-Section Example Cases

Although derivation of the governing equations is not included, results of an evaluation of three, three-section sideload compressors is included to demonstrate the utility of the proposed alternate method of performance analysis. It is evident from an inspection of Figure 9 that six different case examples are included. An odd number of compressor sections results in two different solution approaches to the solution. The specific case designations of "A" and "B" denote these different solution methods. Both solution strategies provide the same results as is evident by equal deviations for all three cases examined.

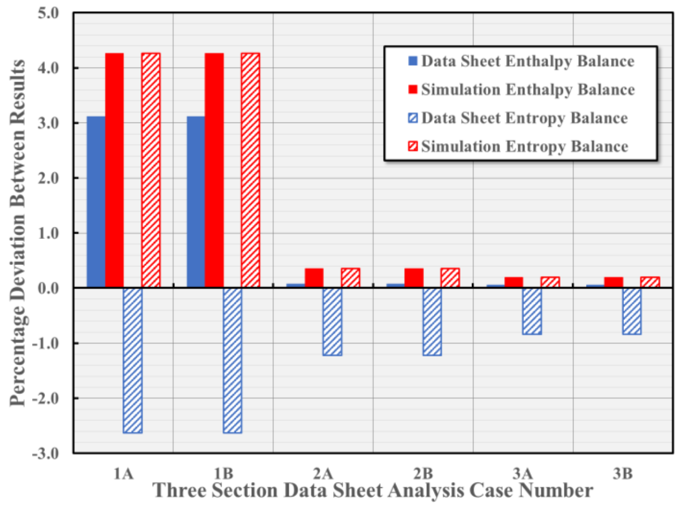

Figure 9. Three-Section Deviation Between Overall and Sectional Energy Balances

These cases include two propane refrigeration compressors. The remaining case (Case 1) is a relatively unique mixed refrigerant application with a substantial portion of the overall compression ratio existing in one of the sections. It is obvious in Figure 9 that both the data sheet and simulation enthalpy balances for this case reflect a significant deviation from sectional to overall balances. The exact origin of these excursions is unclear, but there is some suspicion that a portion of these differences might be due to the equations of state utilized for the data sheet conditions from the validation and simulation method given the complex composition of the mixed refrigerant.

Cases 2 and 3 show good agreement of the enthalpy and entropy balances with maximum deviations on the order of 1 percent or less. Consistent with the two-section cases examined, the data sheet comparisons show that the sectional versus overall entropy balance reflect an underestimate of entropy generation from the overall balances. This may result in an overestimate of sectional efficiency. Simulation enthalpy and entropy balances show equal deviation which is expected with simulation method 1. These deviations are all below 0.5 percent.

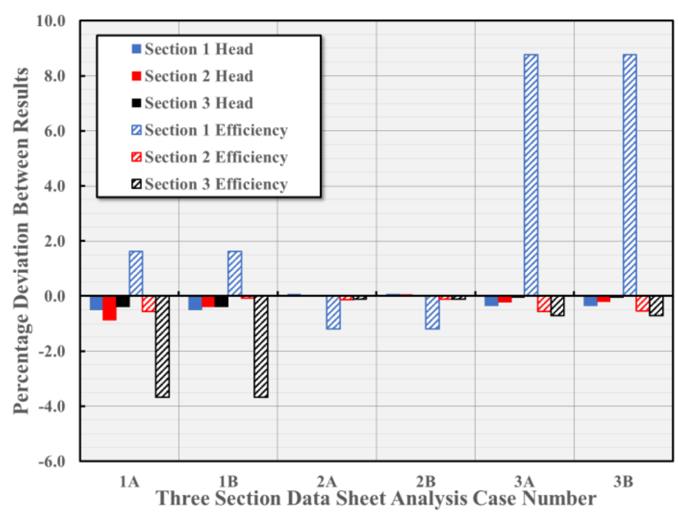

Figure 10. Three-Section Deviation Between Data Sheet and Simulation Polytropic Head and Efficiency

A similar comparison of data sheet and simulation method sectional polytropic head and efficiency is presented in Figure 10. Differentials between the derived polytropic head remain below 1 percent in all three cases for all sections of each sideload compressor case. With the exception of a single section of one case, the differences in sectional efficiencies are below 4 percent with the vast majority of sections below 2 percent. The one section showing an over 8 percent deviation also reflects a simulation predicted discharge temperature nearly 3°F (1.5C) lower than that listed on the related data sheet.

Once again, the results from the simulation method show mostly close agreement with the data sheet parameters notwithstanding some exceptions that are documented with some plausible deviation explanations.

Four-Section Example Cases

The remaining data sheet analyses are included for a four-section sideload compressor configuration. Example cases for three different refrigeration machines are presented with two of the three representing propane compressors. Sectional versus overall enthalpy and entropy balances for each compressor are covered in Figure 11. Exceptional agreement is represented for the enthalpy balances with all deviations within 0.5 percent. Entropy balances are also very favorable, showing agreement at or below 2 percent for all examples. The data sheet entropy balance deviations are mostly negative, again demonstrating a lower amount of predicted entropy generation when compared to the overall balances and the potential to overstate the sectional efficiencies.

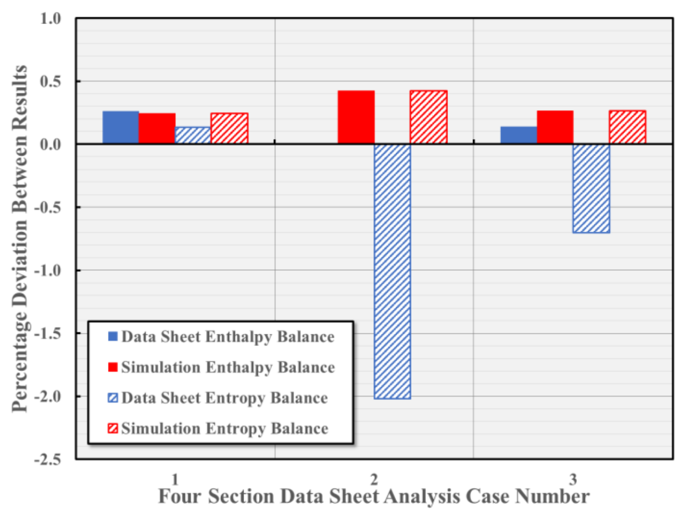

Figure 11. Four-Section Deviation Between Overall and Sectional Energy Balances

Sectional polytropic head and efficiency comparisons display similar trends with data sheet and simulation method derived polytropic head agreeing closely. Efficiencies are within 6 percent also showing a trend of the simulation tending to predict higher sectional efficiencies in the lower stages and lower efficiencies in the higher pressure stages.

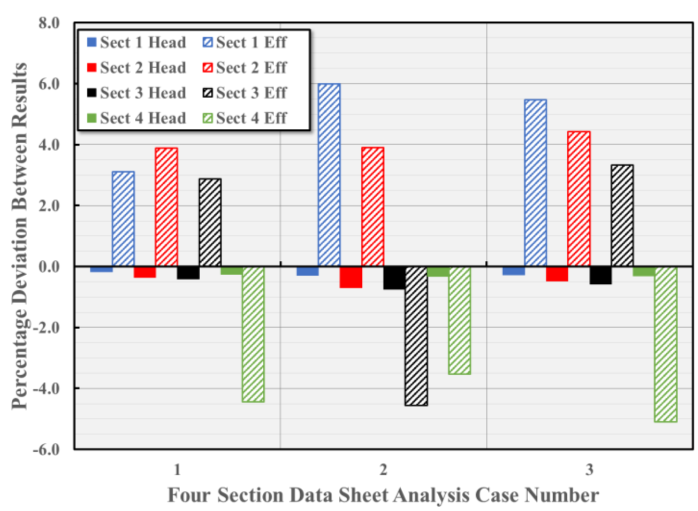

Figure 12. Four-Section Deviation Between Data Sheet and Simulation Polytropic Head and Efficiency

The four-section model is also found to produce reasonably accurate performance prediction when compared to data sheet generated parameters. These analyses have successfully demonstrated the value of the simulation model in predicting sectional compressor performance using only inlet, discharge and sidestream measurable process parameters of total pressure, total temperature and mass flow rate. Independent evaluation or validation of supplier provided performance parameters is possible with the simulation method.

Type 2 Test Evaluation

An alternative approach in the validation of the proposed calculation method for multi-section sideload thermodynamic performance is possible through the evaluation of ASME PTC 10 (1997) Type 2 test data. Results from a factory acceptance Type 2 test of a two-section, variable speed propane refrigeration compressor are presented to illustrate the value of the simulation method. The test gas was a mixture of air and R-134a refrigerant. Multiple test points were recorded at the equivalent design speed along with two additional points each at a lower and higher speed. These off-design speeds are identified as “HS” and “LS,” respectively, for the higher and lower speed conditions.

Sectional curves of normalized, dimensionless performance characteristics are included for the polytropic head coefficient and polytropic efficiency. Predicted performance is depicted as a solid blue curve in each plot. Three sets of different test parameters are also included on each graph. Solid black circle points represent the test data recorded by the equipment supplier and documented in the test report. Red bordered squares denote the supplier test data converted with the NIST REFPROP, Lemmon, et al. (2018) supplied equation of state to ensure consistent results with the simulation data. Finally, the solid red triangles are the calculated results from the proposed simulation method utilizing only the aforementioned externally measured process variables.

First Section Test Data

A plot of the tested polytropic head coefficient versus flow coefficient is shown in Figure 13. Very close agreement is evident among the different calculated values and also the predicted performance. The proximity of the “Test Reported” and “Test Calculated” data supports the premise that the equation of state utilized does not substantially impact the results. Given that the test conditions are near those of an ideal gas, the choice of an equation of state should not influence the results. This may not be the case if the gas conditions deviated more from ideal conditions of pressure and temperature. The close agreement between the “Test Calculated” and “Simulation” data lends validity to the accuracy of the proposed performance determination methodology.

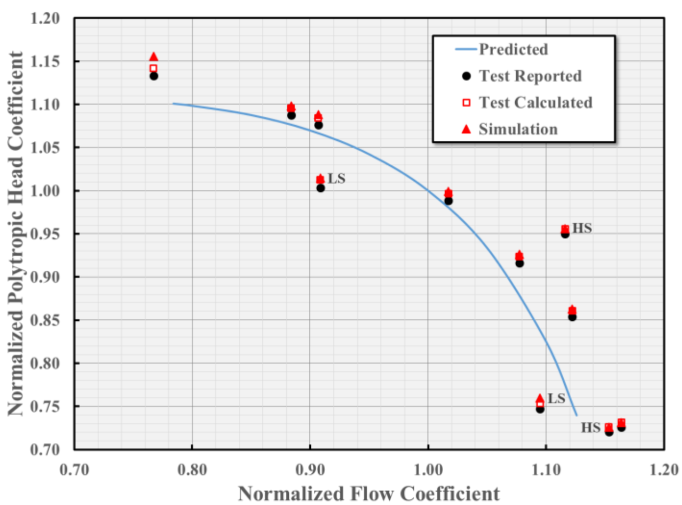

Figure 13. First Section Normalized Head Coefficient versus Flow Coefficient

An examination of the first section polytropic efficiency results in some different observations. While relatively close agreement continues to exist between the efficiencies obtained for the “Test Reported” and “Test Calculated” values, some increased deviation does exist. This is believed to be due to the sensitivity of the entropy and polytropic efficiency to slight variations in temperature. All tested efficiencies calculated for these two data categories are greater than the predicted levels across the entire section flow range. There is a significantly greater difference in the “Simulation” values of the efficiency with most values below the predicted magnitudes except at the high flow coefficient range of the curve.

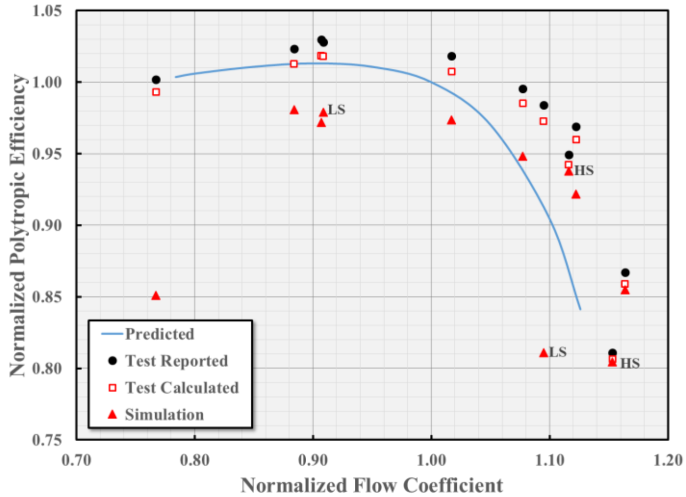

Figure 14. First Section Normalized Efficiency versus Flow Coefficient

The trends exhibited by the first section test results show significant agreement with those of the data sheet analyses presented previously. Specifically, the proposed method closely matches the baseline data sheet and test results for the polytropic head but greater deviations exist when the calculation method is applied to the polytropic efficiency of any given application.

Second Section Test Data

Similar analyses were completed for the second section of the two-section sideload machine that completed Type 2 testing. The results of the test data for the normalized polytropic head coefficient of this second section is provided in Figure 15. Once again, close agreement is shown between the "Test Reported," "Test Calculated" and "Simulation" data points. The tested and method simulated data presents a constant head characteristic above that predicted and also displays an increased flow range over that predicted. These additional consistent results act to further validate the ability of the calculation method to accurately determine sectional polytropic head from only externally available measured process parameters.

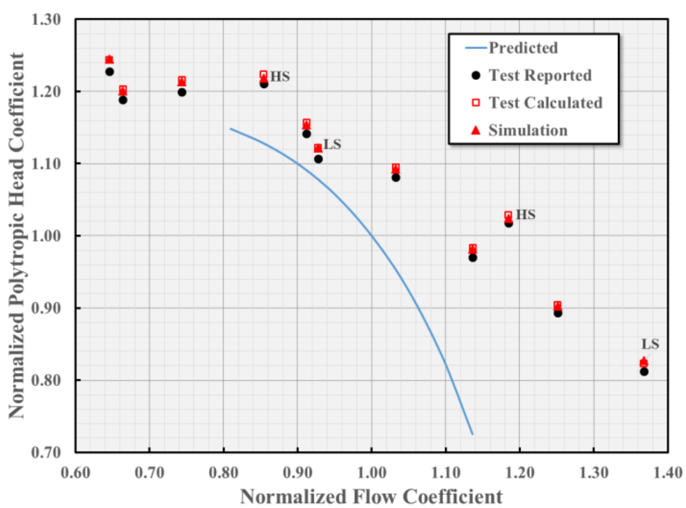

Figure 15. Second Section Normalized Head Coefficient versus Flow Coefficient

Second section test results for polytropic efficiency show equivalent relationships with close agreement between the values reported in the equipment supplier's test report and independently calculated using the measured test variables. There remains a significant difference in the method predicted efficiencies from the test developed numbers with the method predicted values clearly lower than the test derived values. The tested efficiencies tended to reflect higher levels than predicted, whereas those calculated from the simulation method displayed lower than predicted values at the lower end of the section flow range and higher than predicted at higher flows.

A clear conclusion that could be drawn from these results is the value of the proposed method in predicting sectional polytropic head but likely compromised ability to predict section efficiency, however, an examination of energy balances provides some further insight.

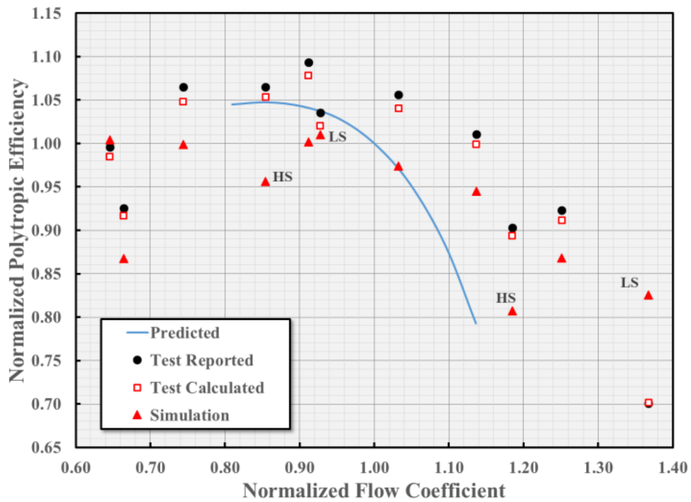

Figure 16. Second Section Normalized Efficiency versus Flow Coefficient

Test Data Overall and Sectional Enthalpy and Entropy Balances
Overall and sectional enthalpy and entropy balances offer some critical insight with the analysis of data sheet performance. This is also true for the Type 2 test data. Examination of the test data enthalpy balance displays a consistent underestimate of the work input by the sectional balances, predominantly less than a 5 percent deviation. The respective comparison of the entropy balances of the test data also shows a significant negative deviation. Notwithstanding the data point identified as 7 DS (which is likely an unsettled surge point), a majority of the data shows negative deviations on the order of 15 to 25 percent. In particular, data points from 2 DS through 5 DS lie within the predicted flow range of the characteristic performance curves. The negative entropy balance deviation is indicative of an estimate of higher sectional polytropic efficiencies than what may actually exist.

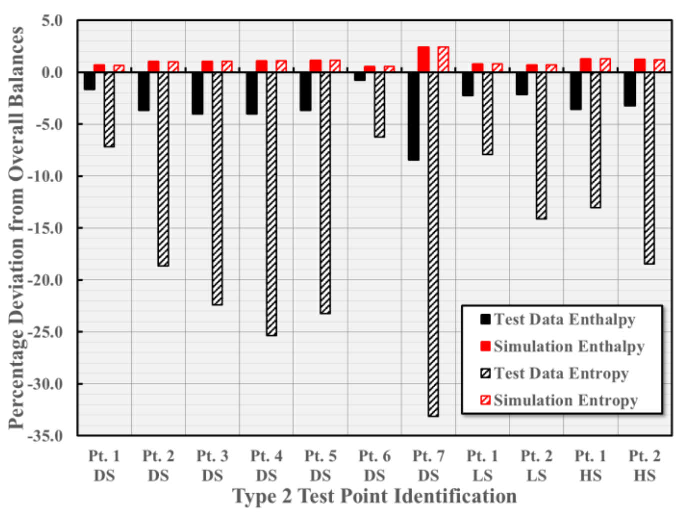

Figure 17. Test Data Overall and Sectional Enthalpy and Entropy Balances

Conversely, the enthalpy and entropy balances of the simulation method all show a positive deviation between the sectional and overall balances. These deviations are all with one exception (point 7 DS) within about 1 percent. Reiterating that method 1 yields equal deviations in the enthalpy and entropy balances and is clearly evident from Figure 17. Application of the other simulation method would result in a perfectly balanced enthalpy comparison, but a larger deviation in entropy. It has also been noted that the differences in sectional polytropic head and efficiency are reduced with the use of method 1. The relatively small deviations in the simulation enthalpy and entropy balances lend further validation of the superior accuracy of the simulation method for test generated performance parameters.

An evaluation of the sectional polytropic head and efficiency derived from the Type 2 test data against the simulation method results calculated from the related test data is presented in Figure 18. It can be observed that the polytropic head of each section generated from the simulation method agrees very closely to that obtained from the Type 2 test data. Deviations are predominantly less than 1 percent with the sole exception associated with point 7 DS, the unsettled surge point. More significant differences are evident with the efficiencies, where the polytropic efficiencies resulting from the Type 2 testing are primarily higher than those determined with the simulation methodology as reflected by the negative deviation values.

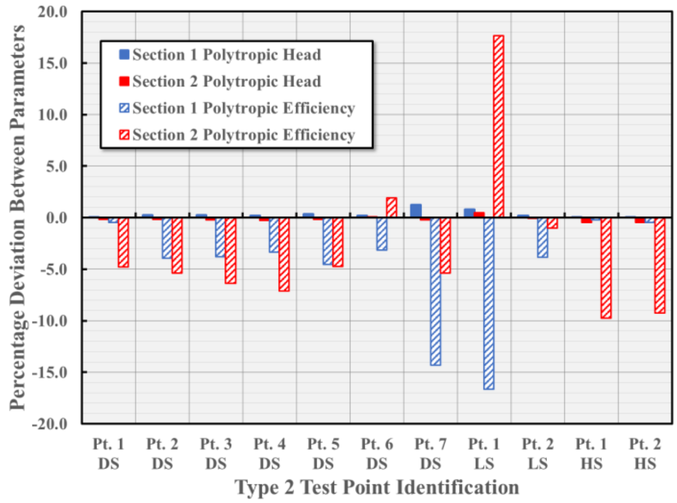

Figure 18. Test Data Sectional Polytropic Head and Efficiency Comparison

In an effort to more completely explain the greater difference in both the entropy balances and magnitude of polytropic efficiency deviations, a comparison of measured and simulation calculated first section discharge and second section inlet temperatures was developed and is plotted in Figure 19. The measured first stage discharge temperature is noted to always be lower than the calculated value leading to a negative deviation for all data points. This results in a higher test efficiency relative to the method calculated value. It was noted previously that the temporary test instrumentation installed to measure first section discharge temperature was subject to error due to the complex and inconsistent flow field in the area of the measurement. Second section inlet temperature should be derived through an enthalpy balance of the first section and sidestream flows. A negative deviation in first stage discharge temperatures is expected to result in a comparable, negative deviation in second section inlet temperatures but this is not the case in the figure.

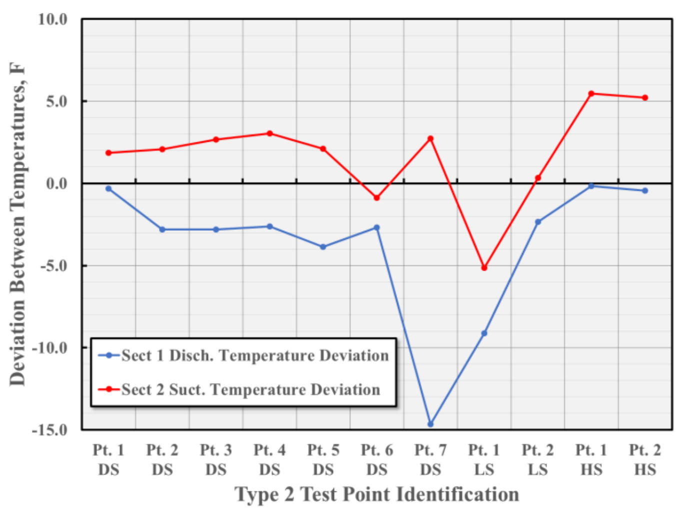

Figure 19. Measured versus Simulated Sectional Temperature Deviation

The exact reason for this discrepancy is not known. A simple enthalpy balance for the stream mixing region as is given in equation 12 supports the expectation that a negative deviation of the second section inlet temperature should be realized in the case of a higher than measured first section discharge temperature. Although this second section inlet temperature is identified as "measured," there is doubt that this was an actually measured value due to complications of the mixing of the two streams of different temperatures in this region. It is possible that the temperature was derived with some enthalpy or temperature mixing calculation with some additional correction factors applied. The net effect of this larger than expected temperature is an overstatement of the second section polytropic efficiency.

Summary

Results of the polytropic head from the simulation method agree very well with both data sheet and actual testing calculations. Comparable numbers for the polytropic efficiency do not agree as closely, but nevertheless provide a reasonable prediction. The bigger question might be the accuracy of the baseline data sheet and test evaluations of polytropic efficiency considering the differential that exists between the overall and sectional entropy balances for a number of the cases provided relative to those calculated from the proposed simulation method. Further investigation of the simulation method developed is warranted.

CONCLUSIONS

A novel method to determine sideload compressor performance only from externally measurable parameters has been developed and introduced. Its ability to accurately calculate internal process and sectional recognized compressor performance parameters has been demonstrated through comparison with both predicted data sheet and actual factory test results. The lack of need to supply an internal temperature (either the first stage discharge or second section inlet) has the potential to enable simpler and more accurate factory acceptance testing and allow longer term performance monitoring once the machine is placed into operation. Although the method has only been presented for a two-section sideload configuration, it can be extended to more numerous sections and results have been included for three and four section examples to illustrate this utility.

While the proposed solution method demonstrates commendable accuracy as presented, a few issues remain that have the potential to further improve the accuracy and enhance its ability to predict sideload compressor internal process and performance parameters.

- The fact that calculation method 1 does not result in a fractional split factor sum equal to unity denotes that some error likely exists. This error, though small (usually less than 1 percent), prevents the overall and sectional balances to converge to an equal value. It is believed that this potential error might be associated with the characterization of the entropy distribution between the sections. Additional effort to improve this aspect of the calculation methodology appears justified.
- It is known that an entropy increase is generated due to the mixing process of two streams with different properties (temperature in this case). Sideload compressor designers attempt to minimize such losses in the design of their sidestream stationary inlet and mixing components, however, it is not known if any entropy generation terms are included in their predicted or tested performance calculations. Further investigation of these mixing design influences and disclosure of their magnitude is recommended.
- Although a number of examples have been provided in order to demonstrate the effectiveness and precision of the proposed methodology, additional and independent verification is suggested in order to further validate and support acceptance.

NOMENCLATURE

h_z = Specific enthalpy at location z
$\dot{m}_z$ = Mass flow rate at location z
P_z = Absolute total (stagnation) pressure at location z
PWR = Gas power
s_z = Specific entropy at location z
Tz = Absolute total (stagnation) temperature at location z
Win = Specific work input based upon total discharge mass flow rate
w_p = Polytropic head (work)
x_z = Mass fraction at location/section z
y_z = Specific work fractional split factor at location/section z
ΔS = Specific entropy generation based upon total discharge mass flow rate
η_p = Polytropic efficiency

Subscripts
1 = Sideload compressor section 1
2 = Sideload compressor section 2
d = Discharge
dz = Discharge at section *z*
dza = Discharge alternate solution at section z
max = Maximum
min = Minimum
s = Suction (or inlet)
ss = Sidestream
sz = Suction (or inlet) at section *z*
sza = Suction alternate solution at section z
t = Total

APPENDIX

A two section propane compressor is evaluated with the following process parameters assumed. The process conditions listed in red font are used in the proposed analysis and would obviously be available from external measurements. Those in blue font are assumed and would not be available from external measurements but would be listed on a data sheet and used to calculate relevant compressor performance parameters. Additional derived thermodynamic properties have been calculated with the NIST REFPROP, Lemmon, et al. (2018), software application for pure propane. The values of enthalpy and entropy in blue font between the sidestream and second section inlet columns were derived from the sidestream balance equations 12 and 13, respectively. Second section inlet enthalpy was also calculated with equation 12 but the entropy was calculated with equation 21. The slight difference in the value of the entropy should be noted and may be the primary influence of the unequal predictions of Method 1 and Method 2.

Location	First Stage		Sidestream		Second Stage	
	Inlet	Discharge			Inlet	Discharge
Pressure (psia):	20.00	70.00	70.00		70.00	245.00
Temperature (F):	-25.0	69.8	37.0		50.19	161.0
Mass Fraction:	0.4		0.6			
Enthalpy (BTU/lbm):	232.04	263.04	249.24	254.76	254.76	289.74
Entropy (BTU/lbm·R):	0.5856	0.5971	0.5702	0.5810	0.5812	0.5949
Specific Volume (ft^3/lbm):	5.0546	1.6826	1.5402		1.5985	0.4939
Compressibility Factor:	0.9556	0.9141	0.8919		0.9017	0.8010

These values are then utilized to evaluate the overall and sectional enthalpy and entropy balances per equations 7 and 8 for the overall balances and equations 10 and 11 for the sectional balances. The percentage difference in these values is also presented in the following table. Another table also demonstrates the calculated polytropic head and efficiency for both sections based upon the parameters provided in the inlet and discharge columns for each section. This information is what might be expected to be included on a data sheet or derived from such information.

Data Sheet Energy Balances			
	Overall	Section	% Difference
Work	47.3805	47.3805	0.0000
Entropy	0.0185	0.0183	-1.1201

Data Sheet Performance		
	Section 1	Section 2
Polytropic Head	19,802.74	21,191.95
Polytropic Efficiency	0.8209	0.7785
Work Input	24,122.07	27,221.22

Utilizing the applicable values necessary to perform the calculations outlined in Table 1, the following values are obtained by iteration to calculate the split factors, y1 and y2, for Method 1 given below in blue font. The iteration limits given by equations 23 through 28 are also included below. As further defined in Table 1, the value for y2 using Method 2 is simple derived by subtracting the value of y1 below from unity.

Enthalpy/Entropy	y1	y2	(y1+y2)
Split Factor	0.2650977	0.7378510	1.0029

	hd1	Td1		
Isentropic S1	257.0264	55.579		
	hs2	Ts2	hd1	Td1
Isentropic S2	261.8656	67.032	280.7997	110.872

y1min	y2min
0.2109239	0.5883752
y1max	y2max
0.4116247	0.7890761

The resulting calculated parameters using Method 1 are provided below. It should be noted that the second section suction pressure is exactly equal to the sidestream and first section discharge pressure as required by the method. The second section inlet temperature and entropy agree very closely to that initially defined. Values and percentage differences in the calculated values of polytropic head and efficiency for each section are also provided based upon Method 1. Finally, a comparison of the overall enthalpy and entropy balances and their percentage differences with the initially defined values are also documented. The equal percentage deviations of overall enthalpy and entropy balances should be noted and is a characteristic of Method 1.

Parameter	Section 1		Section 2	
	Inlet	Discharge	Inlet	Discharge
Pressure (psia):	20.000	70.000	70.000	245.000
Temperature (F):	-25.000	70.747	50.242	161.000
Enthalpy(BTU/lbm):	232.0422	263.4434	254.7834	289.7432
Entropy (BTU/lbm·R):	0.585616	0.597890	0.581241	0.594906
Specific Volume (ft^3/lbm):	5.054605	1.686626	1.598689	1.540198
Compressibility Factor:	0.955630	0.914600	0.901776	0.891946

		Section 1	% Difference	Section 2	% Difference
Polytropic Head (ft-lbf/lbm):		19,826.53	0.1201	21,193.49	0.0073
Polytropic Efficiency:		0.81138	-1.1639	0.77904	0.0682
Work Input (ft-lbf/lbm):		24,435.42	1.2992	27,204.60	-0.0608

		Overall	% Difference
Work (BTU/lbm):		47.5202	0.295
Entropy (BTU/lbm·R):		0.01857	0.295

A set of similar calculated performance parameters are provided below based upon Method 2 from Table 1. In this case, the value of y2 used in the method is simply equal to (1-y1). Deviations in the second section inlet pressure, temperature, enthalpy and entropy from the initially defined values should be noted. Although percentage differences between the polytropic head and efficiency from this analysis and the original values are the same as that in Method 1 for the first section, they are generally more significant for the second section as noted by the increased deviation in the work input. Finally, as dictated by the requirements of Method 2, deviations in overall values of the enthalpy and entropy balances from those defined by the initial data are held to zero.

Parameter	Section 1		Section 2		
	Inlet	Discharge	Inlet	Discharge	
Pressure (psia):	20.000	70.000	70.379	245.000	
Temperature (F):	-25.000	70.747	50.679	161.000	
Enthalpy(BTU/lbm):	232.0422	263.4434	254.9231	289.7432	
Entropy (BTU/lbm·R):	0.585616	0.597890	0.5813	0.594906	
Specific Volume (ft^3/lbm):	5.054605	1.686626	1.590941	0.493873	
Compressibility Factor:	0.955630	0.914600	0.901488	0.801038	

		Section 1	% Difference	Section 2	% Difference
Polytropic Head (ft-lbf/lbm):		19,826.53	0.1201	21,106.47	-0.4033
Polytropic Efficiency:		0.81138	-1.1639	0.77896	0.0572
Work Input (ft-lbf/lbm):		24,435.42	1.2992	27,095.88	-0.4603

		Overall	% Difference
Work (BTU/lbm):		47.3805	0.000
Entropy (BTU/lbm·R):		0.01852	0.000

This example illustrates that the differences between the two methods outlined in Table 1 exist even though they may be slight, however, these small differences may result in more pronounced deviations in derived compressor performance parameters. A good relative indicator of these differences are indicated as the sum of the values of y1 and y2 from Method 1 as compared to unity. As mentioned previously, additional effort is warranted to enhance the method and characterization of the enthalpy and entropy in order to improve the agreement between the two methods. Further investigation to improve the accuracy of thermodynamic property estimation is also a topic of interest since the variation in enthalpy and entropy for a given level of sidestream pressure has been noted to be very small in magnitude.

REFERENCES

ASME PTC 10, 1997, "Performance Test Code on Compressors and Exhausters," American Society of Mechanical Engineers, New York, NY.

Lemmon, E.W., Bell, I.H., Huber, M.L., McLinden, M.O., 2018, *NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP*, Version 10.0, National Institute of Standards and Technology, Standard Reference Data Program, Gaithersburg, MD.

Sandberg, M. R. and Colby, G. M., 2013, "Limitations of ASME PTC 10 in Accurately Evaluating Centrifugal Compressor Thermodynamic Performance," Proceedings of the 42nd Turbomachinery Symposium, Turbomachinery Laboratory, Texas A&M University, College Station, Texas.

Schultz, J. M., January 1962, "The Polytropic Analysis of Centrifugal Compressors," ASME Journal of Engineering for Power, pp. 69-82.

Span, R. and Wagner, W., 1996, "A New Equation of State for Carbon Dioxide Covering the Fluid Region from the Triple-Point Temperature to 1100 K at Pressures up to 800 MPa," J. Phys. Chem. Ref. Data, 25(6):1509-1596.

BIBLIOGRAPHY

Edmister, W. C. and Lee, B. I., 1984, *Applied Hydrocarbon Thermodynamics, Volume 1, Second Edition*, Houston, Texas, Gulf Publishing Company.

Sonntag, R. E. and Van Wylen, G. J., 1991, *Introduction to Thermodynamics – Classical and Statistical*, Third Edition, New York, NY: John Wiley & Sons.

Wark, K., 1995, *Advanced Thermodynamics for Engineers*, New York, New York, McGraw-Hill, Incorporated.

