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DIRECT HEAD MEASUREMENTS FOR TURBOMACHINERY DESIGN

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Brandon Ridens is a research engineer with the Machinery Department at the Southwest Research Institute. He currently oversees the high-pressure gas property testing laboratory at SwRI; providing thermophysical gas properties, including speed of sound, density, specific heat, and enthalpy change measurements for various gases in the high-pressure, supercritical regime. Mr. Ridens has also managed and participated in multiple industry-funded consortia focusing on the thermophysical properties of gases relevant to the compression and pipeline industry and their relation to commonly used equation-of-state (EOS) models. Mr. Ridens has conducted thermodynamic evaluations and reviews of two-phase flow for wet-gas compression and absorption characteristics of CO2, performing thermophysical measurements of supercritical carbon dioxide (sCO2) for compression applications.

ABSTRACT

The isentropic head is one of the most important physical properties for the design and testing of compressors and turbines. It is a critical parameter for the determination of machinery work and power from test data and the associated equipment sizing or matching calculations (Brun and Kurz [2001]). Work, head, isentropic head or polytropic head is usually calculated from an equation of state. Modern equations of state are empirical algorithmic functions or look-up table interpolation expressions of experimental observations that relate primary and directly measured physical properties such as pressure, temperature, density, and speed of sound as well as derived properties such as specific heats, dew point, entropy, enthalpy, etc. To determine enthalpy rise for a process, a path function between two thermodynamic states must be numerically integrated that includes at least three other thermodynamic gas properties such as density, specific heat, and speed of sound, and their derivatives. Other gas property combinations in this path integral are possible but a minimum of three are always required (Coogan et al [2016]). These properties and their derivatives must be known for all points along the integration path to obtain reasonably accurate numerical results for the change in enthalpy.

These gas properties are not directly measured, but are derived from an equation of state prediction, which in itself has the following three sources of uncertainty: (1) uncertainties in the source data or lack of source data, (2) inaccuracy of curve fits, and (3) numerical implementation of thermodynamic equations and mixing rules for gases. Consequently, the combination of source data measurement uncertainties with the derivation uncertainties leads to high equation of state uncertainties for the prediction of enthalpy.

To address the high uncertainties in the determination of enthalpy change, and thus, head, a novel method and apparatus was developed for the direct measurement of this property. This technology was previously described by (Brun et al [2018]) and is based on utilizing a calibrated single stroke near-adiabatic compression event to increase the enthalpy of a test gas between two known thermodynamic states. The approach, while providing reasonably consistent test data for enthalpy, still had a high associated measurement uncertainty because the test apparatus lacked controllability of the piston speed and stroke for all test conditions. Specifically, to truly have repeatable correction factors between a known calibration gas and an unknown test gas requires not only that the test apparatus and test conditions are identical but that the compression stroke in the apparatus has a consistent start position (initial volume), end position (clearance volume), and piston speed. Once that was accomplished, the correction factors described by (Brun et al [2018]) should only be a function of the Reynolds number and can be applied to any complex test gas mixture.

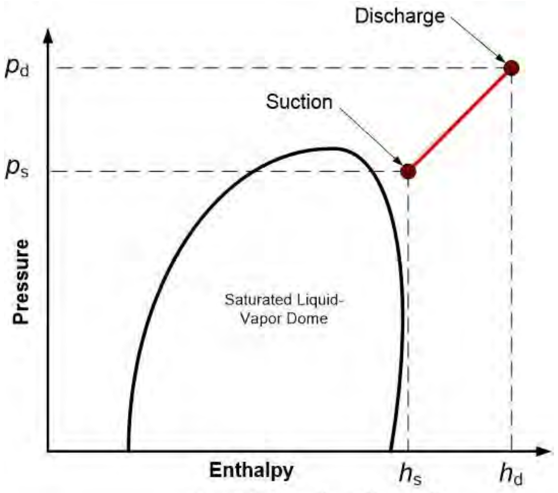
This paper describes the test apparatus modifications that were undertaken to significantly reduce the measurement uncertainties of the head direct measurement apparatus by implementing a consistent single compression stroke using hydraulic actuators and feedback controlled flow valves. The results showed that the isentropic head could be measured for any gas and pressure/temperature state with an uncertainty of less than 0.5%. These test results for enthalpy rise can be used to improve existing equation of state prediction accuracy or directly input to the turbine or compressor design process for any specified application and operating condition. Namely, the developed method and apparatus are the first of its kind practical implementation to reliably and accurately directly measure isentropic head for any gas mixture.

Direct measurement isentropic head test results for two different gas mixtures at each three different pressure/temperature states are presented with their comparison to commonly used equations of state. Results showed that equations of state such as Peng-Robinson, Benedict-Webb-Rubin, and AGA-8/NIST generally accurately predict enthalpy rise at pressures below 2000 psi for typical natural gas mixtures but significant deviations are seen at higher elevated pressures and temperatures. Results also showed that for gas mixtures that contain high percentages of inert components, such as carbon dioxide and nitrogen, or heavy hydrocarbons, commonly used equations of state deviate by more than 3% from direct measurements.

INTRODUCTION

Compressors in oil and gas applications operate at high pressures and moderate temperatures, compressing gas mixtures with a large number of constituents, including carbon dioxide, nitrogen, methane, ethane and other heavier hydrocarbons, as well as, water. Equations of State are used to determine the performance of compressors and pipeline hydraulics. These are semi-empirical models that allow the calculation of thermodynamic and physical properties, such as density, enthalpy, and entropy, of gas mixtures for known pressures and temperatures. While a large body of work is available comparing the results from different equation of state models, very little data is available to verify the results from the various equation of state calculations for the range of pressures, temperatures and gas compositions relevant to many oil & gas operations. This is especially true for dense phase (supercritical) natural gas and gases containing carbon dioxide, hydrogen sulfide, or heavy hydrocarbons. Thus, the users of equations of state (operators and manufacturers) often have no precise data to determine which equation of state gives the most accurate results when predicting compressor performance (Brun et al. [2018]).

When measuring and predicting the performance of centrifugal compressors, the most critical thermodynamic properties from the equations of state are density, speed of sound, entropy, and enthalpy. While density and speed of sound can be measured directly using conventional direct measurement techniques, both entropy and enthalpy are difficult to measure and are usually derived from secondary measurements (Coogan et al [2016]). Therefore, large uncertainties are encountered for these properties in the different equations of state that are commonly used (Ridens et al [2016]). Isentropic enthalpy head (i.e., the enthalpy rise along constant entropy lines) is a critical parameter that is required to accurately design and performance test compressors. Isentropic lines of enthalpy and enthalpy difference are required to determine the power and efficiency of the compression process (Figure 1).



$$H = h(p_d, T_d) - h(p_s, T_s)$$

Figure 1: Definition of Isentropic Head on Pressure-Enthalpy Diagram

Specifically, the most commonly used thermophysical property for machinery performance calculations is enthalpy (h) difference or head ($H=\Delta h$). No direct measurement method for h exists; it must be calculated indirectly from other physical properties. At least three other properties are required to determine head (e.g., $\Delta h = f(v(P,T),a(P,T),c_v(P,T))$). If an equation of state is used only two properties are required but the equation of state requires coefficients for the above values. This results in significant errors that can affect machinery and operational decisions.

A novel method and test apparatus was developed to measure enthalpy rise directly using a calibrated near isentropic compression process (Brun et al [2018]). In this apparatus a test gas is compressed using a fast acting piston inside an adiabatic autoclave. Test results are then corrected using calibration efficiency from a known reference gas compression process at a similar Reynolds number. By using the near isentropic compression cylinder apparatus and the calibration factors (which are determined for a set of reference Reynolds number compression processes) the enthalpy rise of any gas mixture can be accurately determined over a wide pressure and temperature range.

Initial test data showed sometimes unrealistic variations of test data from equation of state predictions. The primary cause of this was that the piston speed of the test apparatus varied widely depending on the test gas and operating pressures and temperatures. Consequently, the test apparatus was improved to implement a fixed piston speed and stroke length for all gas compositions and operating conditions by incorporating a proportional directional control valve. Additionally, various components were modified to allow for accurate measurements at test conditions with pressures up to 17 MPa. This paper describes the test apparatus, the measurement methodology, and the test results for two complex hydrocarbon gases over a range of elevated temperatures and pressures. The uncertainty results are reported and test results are compared to the GERG-2008 equation of state as calculated by NIST's REFPROP program ([Lemmon et al [2013]).

GAS PHYSICAL PROPERTIES AND EQUATIONS OF STATE

Accurate gas property prediction is a necessary component throughout the oil and gas industry for end users, operators, and equipment manufacturers for proper sizing and selection of compressors, improving overall efficiency, and reducing operating costs. The correct determination of the thermodynamic properties of the gas, such as enthalpy, entropy, and specific volume from pressure, temperature, and composition, is imperative for any gas machinery design. Due to the wide range of conditions encountered, pressure, specific volume and temperature (p-v-T) equations of state are used to determine the isentropic or polytropic efficiency, the work input, and capacity of a compressor configuration. Typical equations used include Van der Waal, Lee-Kessler, Lee-Kessler Ploecker, Peng-Robinson, Carnahan-Starling-Desantis, Martin-Hou, Benedict-Webb-Rubin, American Gas Association 8 (AGA8), etc. However, accurate equations of state models are lacking for some of the more complex gas mixtures. Experimentally determined thermodynamic state information is more accurate and is needed to validate, correct, or supplant existing equations of state.

Significant variations between equations of state and deviation from test data have been observed (Ridens et al [2016]). Comparisons of various equations of state versus experimental data for a typical heavy natural gas consisting of 46.5% Methane, 8% Ethane, 5% Propane, 3% I-Butane, 0.5% I-Pentane, 24% Nitrogen, and 5% Carbon Dioxide (all by Mol weight %) resulted in the data shown in Table 1. Even different source implementations of the same equation of state, at the same test condition will give differing results in compressibility, enthalpy, and even molecular weight. For example, for this gas composition the implementation of SRK was compared between PVT Sim, Multi-flash, and also results from REFPROP, Southwest Research Institute's (SwRI®) internal implementation software and other OEM's internal EOS implementation software. Also shown is the significant difference in results between various equations of state for the same gas mixture and test condition. The largest differences, up to 5.6%, were found in calculation results for enthalpy rise (head).

A detailed explanation of equation of state errors is clearly beyond the scope of this paper but the sources can be simplified into the following three categories:

1. Measurement uncertainties in the source data or lack of source data
2. Inaccuracy of curve fits and interpolation approximations from look-up tables
3. Numerical implementation of thermodynamic equations and mixing rules for gases

Any equation of state is only as accurate as the measured data that was used to develop the equation of state. Total measurement uncertainties rely on the individual uncertainties for temperatures, pressures, gas compositions, reference equations of state, data acquisition. Throughout this paper the term "uncertainty" is used. Uncertainty is a statistical absolute probabilistic bound that includes all device, data acquisition, and analysis (non-device) error sources such as precision, bias, readability, hysteresis, etc. Acceptable uncertainties for measurements of speed of sound, density, and specific heat at constant volume near 0.5% can be achieved in a good laboratory. To reach even lower uncertainty requires excessively expensive instrumentation and test procedures which, for economic reasons, is typically not justifiable. The equation of state relies on a mix of curve fits and thermodynamic relationships to interpolate physical properties from the empirical test data. If only limited empirical data is available even the best curve fits will introduce additional uncertainties into the predictions. This added uncertainty is hard to quantify since it depends on the type of curve fit utilized and the implementation of the thermodynamic mixing rules into the model.

Differences between equations of state and between the test data of up to 5% are not uncommon and deviations as large as 10% are sometimes seen. It is difficult to determine whether the sources of these deviations are inaccurate test data or simply the algebraic form or numerical implementation of the equation of state (especially for enthalpy). However, regardless of the source of the inaccuracies in the equation of state, from a compression system design perspective, inaccurate physical property predictions result in engineering analysis mistakes and force over-conservative designs. Furthermore, for the compression system designer it is nearly impossible to know which equation of state will provide the most accurate results for a given gas composition and pressure/temperature range. Direct measurement of gas physical properties is the only way to have a high degree of confidence in the accuracy of the input parameters used in a compressor station design engineering analysis.

The most important physical properties required to analyze compression systems are the fluid's density, speed of sound, enthalpy, and entropy. Density and speed of sound can be readily determined through testing while enthalpy and entropy rely on a complex derivation from other thermodynamic parameters including the gas' specific heat (Coogan et al [2016]). Therefore, developing the capability to measure enthalpy directly provides a significant benefit to the industry.

P1	300	300	300	300	300	300	300	300	300	300	300	300	301.22	301.22	**psia**
P2	1500	1500	1500	1500	1500	1500	1500	1500	1500	1500	1500	1500	1499.1	1499.1	**psia**
T1	120	120	120	120	120	120	120	120	120	120	120	120	119.82	119.82	**Deg F**
T2	387.4	387.2	387	388	388	387	387	387	387	387	389.2	387	389.45	389.45	**Deg F**
Z1	0.952	0.951	0.945	0.959	0.957	0.945	0.959	0.958	0.958	0.959	0.958	0.952	0.947	0.947	
Z1 % Dev.	-0.53	-0.42	0.21	-1.27	-1.06	0.21	-1.27	-1.16	-1.16	-1.27	-1.16	-0.53	0.00	0.00	**%**
Z2	0.965	0.965	0.952	0.984	0.98	0.952	0.995	0.991	0.991	0.996	0.984	1	0.978	0.978	
Z1 % Dev.	1.33	1.33	2.66	-0.61	-0.20	2.66	-1.74	-1.33	-1.33	-1.84	-0.61	-2.25	0.00	0.00	**%**
H1							57.8	57.66	57.68	20.248					**kJ/kg**
H2							300.71	300.08	300.12	261.19					**kJ/kg**
Head	238.7	238.9	236.3	241.8	240.5	236.3	242.9	242.4	242.4	240.9	241.8		250.3	251.1	**kJ/kg**
Head % Dev	4.63	4.55	5.59	3.40	3.92	5.59	2.96	3.16	3.16	3.76	3.40		0.00	-0.32	**%**
Molecular Weight	25.826	25.826	25.826	25.826	25.826	25.826	25.825	25.825	25.825	25.825	25.825	25.819	Assumed (25.825)	Assumed (25.825)	
	RK (OEM)	*SRK (OEM)*	*PR (OEM)*	*LKP (OEM)*	*BWRS (OEM)*	*SRK (OEM)*	*SRK (PVTSim)*	*SRK Peneloux (PVTSim)*	*SRK Peneloux (T) (PVTSim)*	*Multi-Flash*	*Refprop*	*RK (SwRI)*	*Tested (Refprop EOS Correction)*	*Tested (SRK EOS Correction)*	

Table 1. Deviations of Compressor Performance and Physical Properties for Various EOS

ENTHALPY

The basic properties of pressure, temperature, and specific volume (or density) are fundamentally defined by how they are measured, and can be directly observed by experiment. Other quantities involving the derivatives of thermodynamic properties can also be measured, such as the speed of sound and the specific heat at constant volume. Conversely, enthalpy, while just as fundamental as the basic properties, is a derived quantity that conventionally is not directly measured. This crucial property is usually calculated from the experimentally measurable quantities of pressure, temperature, specific volume, speed of sound, and specific heat at constant volume (Coogan et al [2016]). Alternatively, it can be calculated using specific heat at constant pressure for an ideal gas and the departure function to account for non-ideal gas behavior. Specifically, Table 2 shows how enthalpy is calculated from primary measurement properties. All integrals and partial derivatives are in a form that can be computed numerically from the experimental data. One should note that since the integrals are path dependent, experimental data for numerically integration must be available along the entire compression path, not just at the starting and end points.

Ind. Variables	Enthalpy Calculation
T, P	$\Delta h_{12} = \int_{P_1}^{P_2} \left[v - T\left(\frac{\partial v}{\partial T}\right)_P \right]_{T=T1} dP + \int_{T_1}^{T_2} \left[-c_v \frac{a^2}{v^2} \left(\frac{\partial v}{\partial P}\right)_T \right]_{P=P_2} dT$
T, v	$\Delta h_{12} = \int_{T_1}^{T_2} \left[c_v + v\left(\frac{\partial P}{\partial T}\right)_v \right]_{v=v_1} dT + \int_{v_1}^{v_2} \left[T\left(\frac{\partial P}{\partial T}\right)_v + v\left(\frac{\partial P}{\partial v}\right)_T \right]_{T=T_2} dv$
P, v	$\Delta h_{12} = \int_{P_1}^{P_2} \left[c_v \left(\frac{\partial T}{\partial P}\right)_v + v \right]_{v=v_1} dP + \int_{v_1}^{v_2} \left[c_v \frac{a^2}{v^2} \left(\frac{\partial T}{\partial P}\right)_v \right]_{P=P_2} dv$

Table 2. Final Results – Enthalpy

Figure 2 shows the path integral required to determine isentropic head between any two state points. It is a path integral using density, speed of sound, and specific heat and each individual property has high EOS uncertainty by itself.

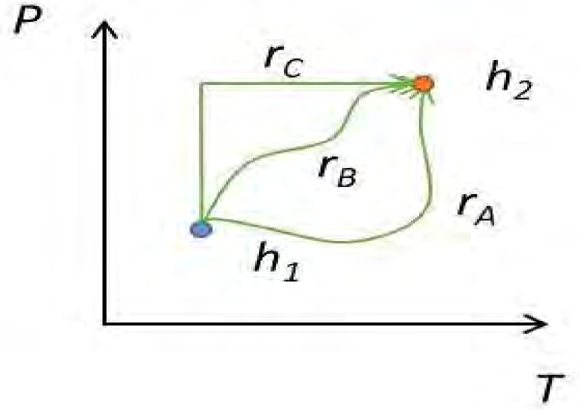


Figure 2: Path Integral to Determine Enthalpy Rise Between Two States

METHODOLOGY TO DETERMINE ENTHALPY RISE

A novel method and test apparatus was developed to measure enthalpy rise directly using a calibrated near isentropic compression process. In this apparatus a test gas is compressed using a fast acting volume change (either using a piston or bladder) inside an adiabatic autoclave. Figure 3 shows a conceptual schematic of the apparatus. Test results are then corrected using a calibration efficiency from a known reference gas compression process at a similar Reynolds number. By using the near isentropic compression cylinder apparatus and the calibration factors (which are determined for a set of reference Reynolds number compression processes) the enthalpy rise of any gas mixture can be accurately determined over a wide pressure and temperature range.

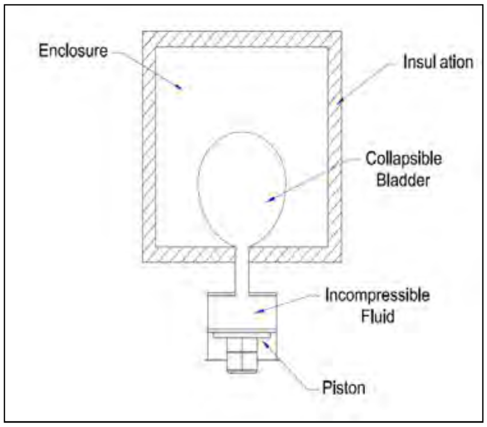


Figure 3. Schematic of Apparatus to Test

The basic analysis and test approach relies on completing two identical near adiabatic compression processes for both the known calibration gas and the unknown test gas. Within each compression process work is added to the fluid inside the adiabatic enclosure via a rapid volume change of the bladder:

$$W = m \triangle h \qquad \textit{Equation 1}$$

where:

$$W = \int \boldsymbol{VdP} \qquad \textit{Equation 2}$$

Thus, the actual enthalpy rise of the gas in the enclosure is simply:

$$\triangle h = \frac{\int \boldsymbol{VdP}}{\boldsymbol{m}} \qquad \textbf{Equation 3}$$

or:

$$\Delta \boldsymbol{h} = \frac{\int \boldsymbol{VdP}}{\boldsymbol{V_1 \rho_1}} \qquad \textbf{Equation 4}$$

Where V_1 and ρ_1 are the initial volume and density of the process gas in the enclosure. However, even if the compression process is near isentropic (adiabatic and fully reversible) it can never be 100% efficient. Thus, a compression efficiency η is defined as:

$$\eta = \frac{\Delta h_{isen}}{\Delta h_{actual}} \qquad \textbf{Equation 5}$$

Since the volume of the enclosure is difficult to accurately and repeatedly determine, it is convenient to eliminate volume as a free variable and define a new volumetric efficiency $\widetilde{V}$ that includes the initial volume of the compression process:

$$\widetilde{\boldsymbol{V}} = \frac{\boldsymbol{V_1}}{\boldsymbol{\eta}} \qquad \textbf{Equation 6}$$

Thus, the isentropic enthalpy difference (isentropic head) of the compression process is:

$$\boldsymbol{\Delta h_{isen} = \eta \, \Delta h_{actual} = \eta \, \frac{\int Vdp}{V_1 \rho_1} = \frac{\int Vdp}{\widetilde{V} \rho_1}} \qquad \textbf{Equation 7}$$

$\tilde{V}$ (calibration volume) corrects for the inefficiency of the test gas compression process. $\tilde{V}$ is obtained from calibration testing using a reference gas with a known enthalpy rise and ρ_1.

$$\widetilde{\boldsymbol{V}} = \frac{\boldsymbol{\Delta h_{isen} \rho_1}}{\int \boldsymbol{Vdp}} \qquad \textbf{Equation 8}$$

Nitrogen or carbon dioxide can be utilized for calibration testing since the enthalpies for these gases are available from credible EOS sources with a demonstrated low uncertainty (e.g., NIST Refprop). Typical enthalpy measurement uncertainties of these calibration gases are well below 0.1%. Calibration testing was performed to obtain $\widetilde{\boldsymbol{V}}$ for a range of Reynolds numbers. Here the Reynolds number is defined using piston diameter for length, piston velocity for speed, and initial gas properties for density and viscosity. Also, technically the calibration volume is also a function of the heat transfer between the gas to the cylinder during the compression process. To minimize this non-linear and hard to capture effect, the cylinder is internally insulated to minimize heat transfer, the compression stroke action is very short, and the initial conditions of the test gas and that of the calibration gas are at similar temperatures. The test error due to heat transfer was estimated to be less than 0.1%.

UNCERTAINTIES OF PHYSICAL PROPERTY MEASUREMENTS AND ENTHALPY CALCULATION

The test uncertainty was calculated for the primary direct measurements and reference condition (pressure, temperature, EOS model prediction, and gas mixture uncertainty) at each test condition. The test uncertainty is a function of the sensor measurement uncertainty, gas composition uncertainty, geometric uncertainties, reference equation of state uncertainty, and data acquisition uncertainties. The instrument uncertainty estimate is described for the pressure, temperature, density, and enthalpy measurement below.

Prior to determining a test uncertainty, it is important to know whether the measured variables in the test are independent or dependent. This aids in determining which method of calculating uncertainty must be employed. If the measured variables in an experiment are truly found to be independent, then the method to determine total uncertainty is simply an addition of all individual measurement uncertainties. However, in the case of enthalpy testing the direct measurements are pressure, temperature, and volume change which are functionally dependent parameters. The most accurate analysis to determine total uncertainty of dependent variable measurement systems is the perturbation method, which is based on the actual input function and does not require any linearity assumptions (Brun and Kurz [2001]).

The perturbation method is implemented by sequentially perturbing the input values, such as temperature and pressure, by their respective uncertainties and recording the effects on the calculated output quantity (enthalpy). Using the perturbation method, the influence of each test parameter uncertainty can be directly evaluated for the particular gas mixture at the pressure and temperature condition selected for the iteration. The NIST REFPROP (Lemmon et al [2013]) program is used for all EOS model predictions and to evaluate the gas mixture uncertainty, pressure uncertainty, volume uncertainty, and temperature measurement uncertainty on the enthalpy. To find the final test uncertainty, the root sum square addition is used to represent a true superposition of all positive and negative uncertainties.

The uncertainty in each gas component was reported to be between 1%-2% based on the certification supplied by the gas supplier. This effect was modeled in the NIST REFPROP software by considering the analytical uncertainty effect on the primary constituent. This method used allowed the uncertainty to be customized for the specific gas mixture and pressure/temperature condition of interest.

The instrument uncertainties, referenced as condition uncertainty, are uncertainties associated with the state point conditions (pressure, temperature, EOS model prediction, and gas mixture uncertainty) that are used as references when comparing the measurement to EOS. The reference uncertainty is highly dependent on the EOS used and varies with gas composition. The reference condition uncertainty varied from 0.07% to 0.83% depending on the test condition. The driving factor for the larger uncertainties in the reference condition is typically attributed to either the temperature or mixture component uncertainties depending on the specific mixture and test point (their sensitivity in influencing the predicted property from the EOS).

Tests were repeated a number of times to obtain a reasonable sample of measurement points. The total measurement uncertainties were calculated and compared to test sample repeatability to make sure that test variations fall within the total calculated uncertainties. Total measurement and data acquisition uncertainties were predicted using the perturbation method and validated versus data scatter. Typical uncertainties for the total uncertainty band around a single test point are test specific but usually ranged between 0.1-1.2% of the enthalpy difference. The calculated uncertainty bands are included in the measured test results and charts.

Uncertainty calculations for the calculation of Enthalpy from the measured data were performed using Python “Uncertainties” which calculates the standard deviation of mathematical expressions through the first order linear approximation of error propagation theory. Interpretation of the uncertainty results must be performed with the understanding that the uncertainty values are probability distributions with an interval size approximately the size of one standard deviation, and the nominal values are the locations of highest probability.

For each data set, uncertainty of the VdP integral is calculated, and an estimated uncertainty for the calibration term, initial density, and specific heat ratio is used. The estimated percent error of these values is 1% because all of the instrumentation measurements used to calculate them had uncertainties less than 1%. A 1st degree polynomial (trapezoidal rule) was used for the uncertainty calculations. Total measurement uncertainty was found to be less than 2% for all data points as shown in Table 3 for Gas Composition 1 and Table 4 for Gas Composition 2.

Start Temperature (T1)	**Start Pressure (P1)**	**Uncertainty - Condition**	**Uncertainty - Measurement**
deg C	**MPa**	**%**	**%**
26.148	2.089	0.542	1.227
27.793	3.450	0.679	1.227
106.449	6.220	0.585	1.227
27.124	7.617	0.065	1.227
100.493	10.374	0.657	1.227
98.583	13.099	0.280	1.228
97.636	15.872	0.108	1.230

Table 3. Uncertainty Calculations - Gas Composition 1 (GMRC Mix 2)

Start Temperature (T1)	**Start Pressure (P1)**	**Uncertainty – Condition**	**Uncertainty - Measurement**
deg C	**MPa**	**%**	**%**
27.506	2.075	0.635	1.227
27.936	3.454	0.833	1.227
99.356	6.226	0.736	1.228
27.588	7.610	0.400	1.228
102.166	10.322	0.728	1.227
89.835	13.223	0.400	1.228
89.293	16.027	0.115	1.231

Table 4. Uncertainty Calculations – Gas Composition 2 (PRCI Mix 2)

ENTHALPY TESTING

MODIFIED TEST SETUP

The original test device designed to measure the enthalpy rise in a specified test gas during a repeatable compression event were modified to improve the accuracy and repeatability of the process. The device was modified by replacing the solenoid valve from the previous configuration (See Figure 4) with a Parker proportional directional control valve controlled with a programmable RMC70 motion controller; replacing the accumulator with a larger volume bladder dampener; and installing a new pump with a larger flow capacity and higher head. The first benefit of these modifications was to create a consistent compression process using a constant fixed piston speed for all gas compositions and operating points tested, i.e. the piston speed is the same for compressing a gas at 0.1 MPa (14.7 psia) as well as at 15.9 MPa (2,300 psia). Another improvement from using the control valve is that it allowed for precise control of the stroke length which created a constant start and stop position. This was first attempted by using a hard stop for the start position of the piston; however, it was found in the initial testing that some of the uncertainty in the results was due to the piston "floating" against the hard stop position thus introducing significant uncertainty in the fixed initial volume measurement. Finally, increasing the size of the accumulator as well as the pump increased the range of the device to accurately test gas compositions with initial start pressures of up to approximately 17 MPa (2,500 psia).

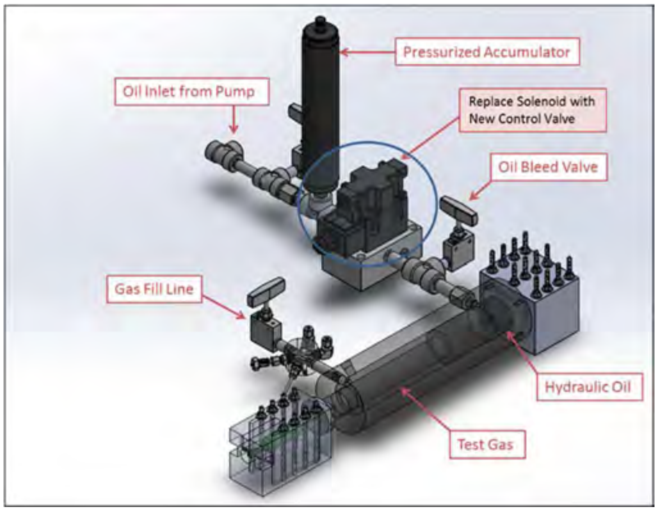


Figure 4. Solid Model of Modified Enthalpy Rise Test Apparatus

As originally designed, the compression event was created using a hydraulics system and a hydraulic cylinder (See Figure 4). The rod-side of the hydraulic cylinder was filled with the test gas at the initial test conditions. A manifold was connected to the rod-side port to accommodate all instrumentation connections, as well as, the gas fill line. The other side of the hydraulic cylinder was filled with hydraulic oil. The oil-side port was connected to the control valve and pressurized accumulator. Actuating the control valve allowed the pressurized oil stored in the accumulator to rapidly move the piston in the hydraulic cylinder from the initial position to the final position, compressing the gas. A fast but fixed compression process for this testing is preferred since it: (1) closely replicate the actual conditions in a natural gas compression system, and (2) reduces the heat transfer loses and their non-linear effects on the test results. With the control valve installed, the compression stroke time from the device was measured at approximately 0.242 seconds with data collected over pressures ranging from 1.4 MPa-10.5 MPa (200-1520 psi) at ambient temperature. The constant piston velocity and stroke length over the range of tests for CO_2 is shown in Figure 5.

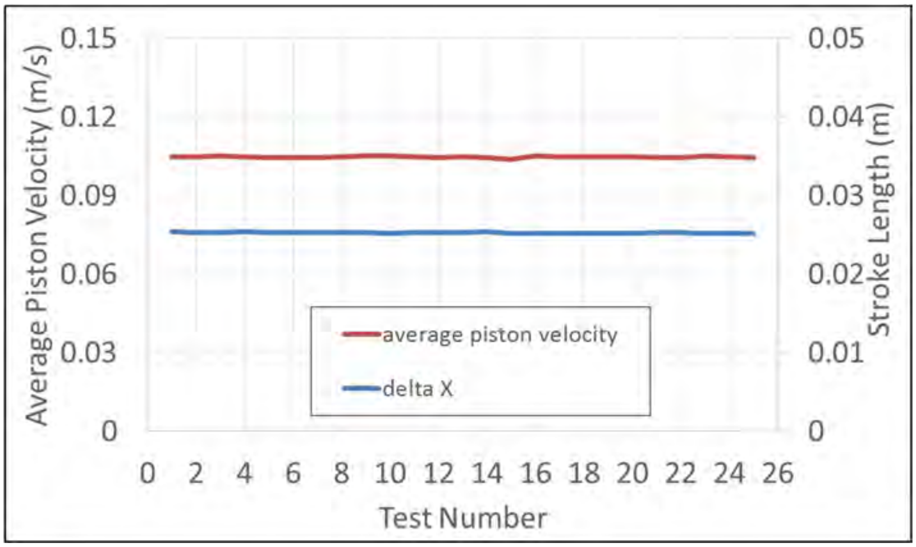


Figure 5. Piston Velocity and Stroke Length of Compression Process with New Control Valve

In order to calculate the enthalpy rise in the test gas (see the equations in Section 5), the following information must be known: transient pressure and volume of the gas during the compression event as well as the mass of gas to be compressed.

The data was recorded using a custom LabVIEW interface that reads multiple data acquisition modules. Individual modules to read pressure, temperature and voltage were installed on one chassis to a single computer so that multiple signals could be simultaneous recorded (synchronized) with the same timestamp and frequency.

PRESSURE AND TEMPERATURE MEASUREMENTS

A Honeywell Sensotech static pressure transducer was used for the primary test gas pressure measurement. The specified accuracy of this sensor is 0.10%. During testing, this sensor proved to have a fast enough response time to be used for the numeric integral calculations.

A PCB dynamic pressure transducer was also installed to measure the test gas pressure. The non-linearity of this sensor is specified as 1.0%. The dynamic pressure transducer was able to capture the full pressure increase during compression for most tests, as the compression time was very fast. The dynamic pressure measurements were used to validate the static pressure readings.

A Nanmac C2-7, K-type ribbon thermocouple (slow thermal response) was installed to measure the test gas initial temperature. The tolerance value was specified as 1.1 °C or 0.4% of the temperature range (0-750 °C), whichever is greater. The accuracy of the thermocouple measurement was validated with an RTD (See Figure 6). For the test points where the required initial gas temperature was above ambient temperatures the hydraulic cylinder was wrapped with heat tracing and an external temperature controller was used to heat the cylinder and test gas to the desired temperature.

VOLUME MEASUREMENT

The change in volume during the compression event was measured using a displacement sensor connected to the hydraulic cylinder push rod (See Figure 6). The volume was then calculated as the displacement multiplied by the cross sectional area of the hydraulic cylinder.

The displacement sensor installed is a linear potentiometer position sensor (LPPS) from HGSI. The non-linearity of this sensor is specified as 0.06% of the full stroke. The repeatability of sensor measurements is specified at 0.01 millimeters. The maximum operating speed specified for the sensor is approximately 4.8 m/s.

The displacement sensor was also used to ensure repeatability in the compression processes for the enthalpy rise test assembly. The compression stroke, i.e., the initial and final positions of the piston (and therefore the initial and final gas volumes) must be consistent for each test. The initial piston position was verified during each test using the displacement sensor output. For each test, the final position was full extension of the piston rod.

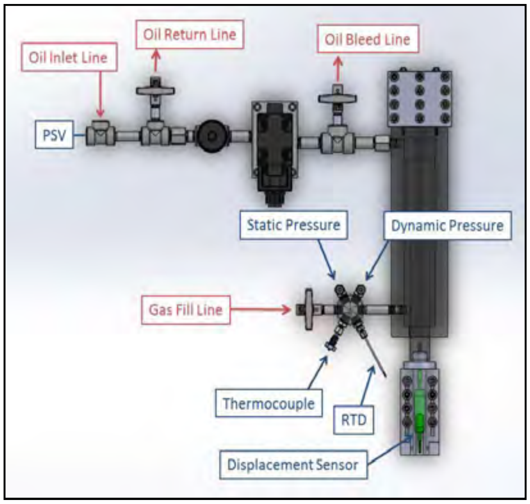


Figure 6. Solid Model of Enthalpy Rise Test Apparatus with Instrumentation Locations Labelled

TEST PROCEDURE

The enthalpy rise apparatus was assembled and tested in a dedicated gas properties testing laboratory. The test assembly was connected to the hydraulic pump system using high-pressure flexible hoses. The accumulator was pre-charged with Nitrogen to 17.2 MPa (2,500 psi). A picture of the test assembly is shown in Figure 7.

Pressure, temperature, and displacement data were monitored continuously using the NI DAQ system during all setup and testing procedures. The RMC70 motion controller was programmed to control the piston speed and initial and final positions (stroke length) at fixed values with the use of a control valve. Before pressurizing with test gas, the piston rod was fully extended and a vacuum was pulled on the gas volume and all hoses to remove any air. The gas side of the hydraulic cylinder was then filled with the specified test gas and the piston was retracted to the initial position by redirecting oil flow through the control valve. The gas side of the apparatus was then pressurized to the required initial pressure. For the test points where the required initial gas temperature was above ambient temperatures the hydraulic cylinder and tubing were wrapped with heat tracing and an external temperature controller was used to heat the cylinder and test gas to the desired temperature.

Once the test gas reached the required initial pressure and temperature, the accumulator was pressurized using the hydraulic pump and the initial position of the piston was verified using the displacement sensor. The control valve was then actuated to initiate the compression event. Transient pressures and displacement data were recorded during the test. These compression tests were repeated three to five times for each test point to obtain a reasonable sample size per test point.

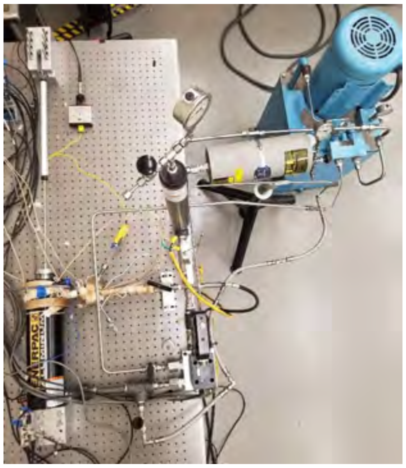


Figure 7. Enthalpy Rise Apparatus Assembly, Top View (Uninsulated)

RESULTS AND DISCUSSION

DEVICE CALIBRATION

The initial testing of the modified enthalpy rise apparatus was conducted using nitrogen and carbon dioxide as a calibration gases. Nitrogen and carbon dioxide have well documented physical properties and accurate data is available in the public domain from several test validated sources. Twenty-five test points were taken for the calibration testing with an example of ten of the initial points for tests conducted for Nitrogen summarized in Table 5. Data was recorded for several iterations of each initial test point. Similar testing was performed with carbon dioxide.

Test Point	Initial Pressure MPa [PSI]	Initial Temperature °C [°F]
1	1.7 [245]	25 [77]
2	2.1 [300]	25 [77]
3	2.4 [350]	25 [77]
4	3.1 [450]	25 [77]
5	3.4 [500]	25 [77]
6	3.8 [550]	25 [77]
7	4.2 [600]	25 [77]
8	4.5 [650]	25 [77]
9	4.8 [700]	25 [77]
10	5.0 [730]	25 [77]

Table 5. Nitrogen Test Points Used for Enthalpy Rise Device Calibration

The pressure, temperature, and displacement data recorded during the calibration testing were used to calculate the actual enthalpy rise of the gas using $\triangle h = \int VdP/V_1P_1$using **Equation 4**. This calculated enthalpy rise was then compared to the isentropic predicted value using the default NIST equations of the thermodynamic and transport property software REFPROP.

The measured and the equation of state enthalpy rise were used to calculate the calibration volume $\tilde{V}$, for each calibration test from $V = \Delta h_{isen}\ \rho 1Vdp$ **Equation 8**. The measured data was also used to calculate the Reynold's number for each test point. Figure 8 shows a plot of the calibration volume $\tilde{V}$, versus the Reynold's number. As expected, with a constant piston velocity and stroke length for all test points, the calibration line fit to the data points has a relatively flat slope for the range of relevant Reynold's numbers. For subsequent tests, this calibration line was used to determine enthalpy rise of test gases.

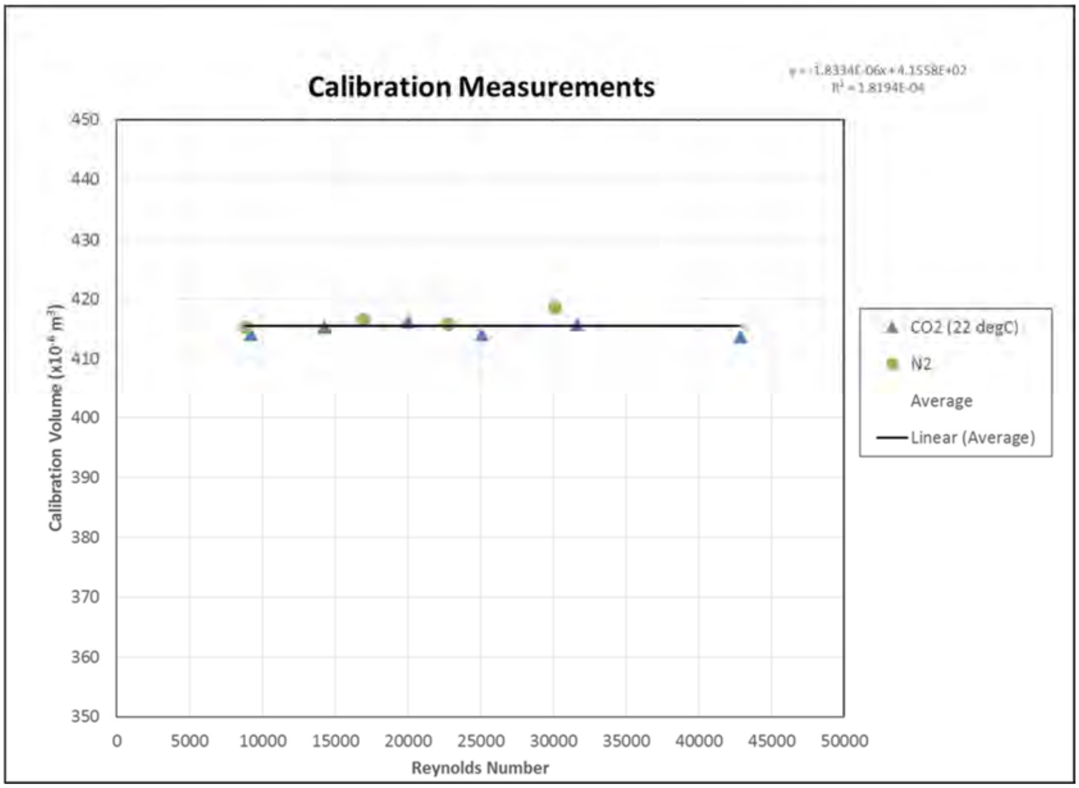


Figure 8. Calibration for Enthalpy Rise Device Derived Using Nitrogen and Carbon Dioxide

GAS COMPOSITIONS AND TEST POINTS

The test points that were measured are shown in Tables 6 through 9. These points represent conditions typically encountered for the previously tested mixtures defined in previous projects (Brun et al [2018]) that measured the density, speed of sound and specific heat at a constant volume. However, thermophysical property predictions for these gases are not well known for higher pressures, particularly near the supercritical region.

The first gas composition is shown in Table 6 and is identical to the gas composition labeled GMRC Mix 2 from the Equation of State Testing Project. This gas has similar amounts of CO_2 and Methane with 20% heavier hydrocarbon and nitrogen components. The test points are shown in Table 7 and are plotted in relation to the critical point on the gas mixture's pressure-enthalpy (P-h) phase diagram shown in Figure 9.

Actual	%mol
CO_2	39.99
Methane	41.002
Ethane	9.999
Propane	5.002
Butane	1
Nitrogen	3.007

Table 6. Gas Composition 1 (GMRC Mix 2)

Temperature C (F)	Pressure MPa (psia)	Enthalpy (kJ/kg)	Entropy (kJ/kg-K)	Quality	Phase
27 (80)	2.1 (300)	596.1	3.24	Superheated	Gas
106 (223)	6.2 (900)	687.9	3.24	Superheated	Gas
27 (80)	3.4 (500)	580.5	3.07	Superheated	Gas
109 (229)	10.3 (1500)	667.5	3.07	Undefined	Supercritical
27 (80)	7.6 (1100)	524.8	2.72	Superheated	Gas
99 (210)	13.1 (1900)	630.1	2.92	Undefined	Supercritical
100 (212)	15.9 (2300)	616.2	2.84	Undefined	Supercritical

Table 7. Gas Composition 1 Test Points and Phase (GMRC Mix 2)

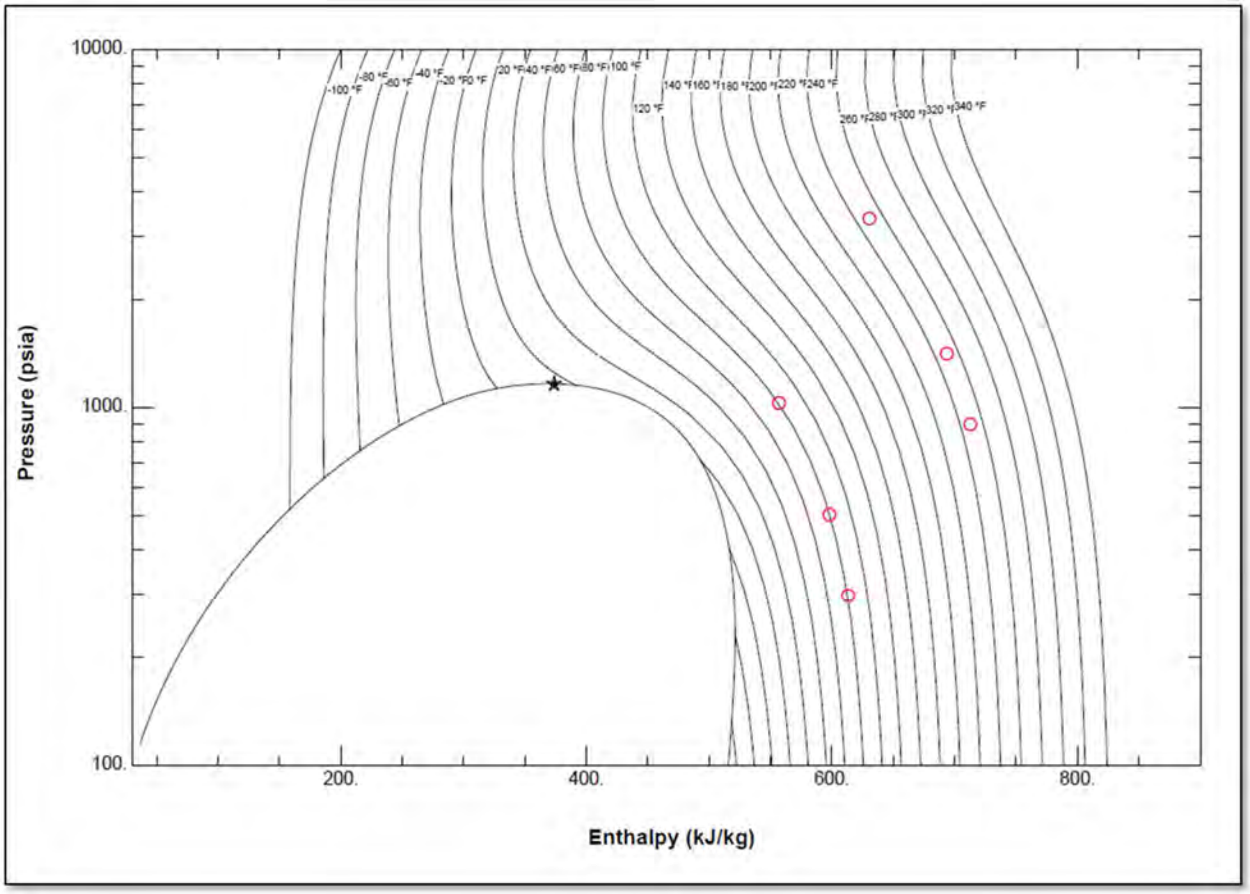


Figure 9. Gas Composition 1 Test Points Plotted on Phase Diagram

The second gas composition is shown in Table 8 and is identical to the gas compostion labeled GMRC Mix 2 from the Equation of State Testing Project. This gas has relatively high amounts of heavier hydrocarbons and was originally selected by PRCI due to test data in the field not matching thermodynamic predictions from EOS. The test points are shown in Table 9 and are plotted in relation to the critical point on the gas mixture's pressure-enthalpy (P-h) phase diagram shown in Figure 10.

Actual	%mol
CO_2	2.003
Methane	69.6059
Ethane	15
Propane	10
i-Butane	0.9999
Butane	0.9994
i-Pentane	0.1501
Pentane	0.15
Hexane	0.1
Nitrogen	**0.9917**

Table 8. Gas Composition 2 (PRCI Mix 2)

Temperature C (F)	Pressure MPa (psia)	Enthalpy (kJ/kg)	Entropy (kJ/kg-K)	Quality	Phase
27 (80)	2.1 (300)	738.3	4.05	Superheated	Gas
100 (212)	6.2 (900)	861.6	4.05	Superheated	Gas
27 (80)	3.4 (500)	717.7	3.82	Superheated	Gas
103 (218)	10.3 (1500)	834.9	3.82	Undefined	Supercritical
27 (80)	7.6 (1100)	647.7	3.35	Superheated	Gas
90 (194)	13.1 (1900)	774.1	3.58	Undefined	Supercritical
100 (212)	15.9 (2300)	783.8	3.54	Undefined	Supercritical

Table 9. Gas Composition 2 Test Points and Phase (PRCI Mix 2)

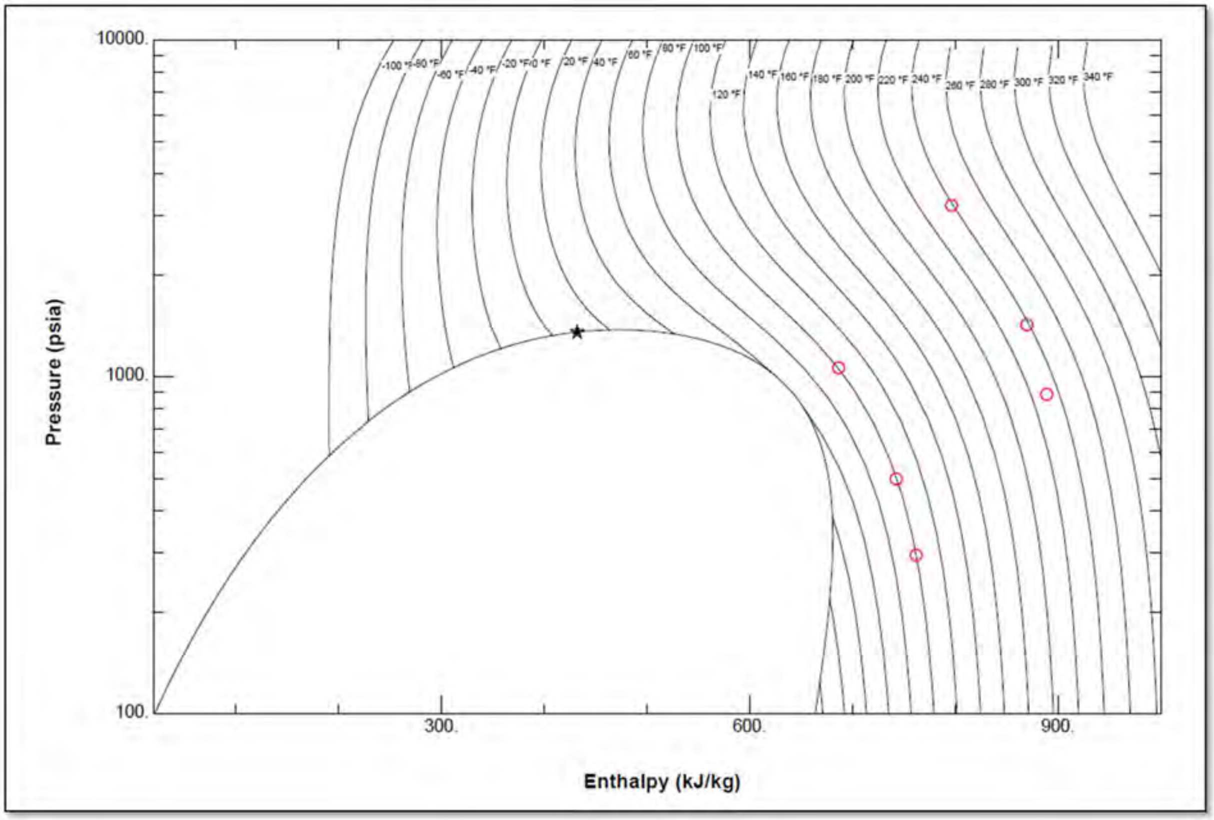


Figure 10. Gas Composition 2 Test Points Plotted on Phase Diagram

ENTHALPY RISE TESTING

The updated test apparatus was used to measure the enthalpy rise of the two gas mixtures. Transient pressure, temperature, and displacement data were recorded during the compression of the test gas from these initial conditions. The Reynold's number was also calculated for each test and used to determine the appropriate calibration volume value from the calibration line (See Figure 11). The calibration volume and the measured data were then used to calculate the actual enthalpy rise in the test gas during compression using $\Delta h_{isen} = \eta\ \Delta h_{actual} = \eta\ \ VdpV1\ \rho 1\ ) = VdpV\rho 1$ Equation 7. Figure 11 shows the calculated enthalpy rise values plotted versus the corresponding enthalpy rise values predicted by the NIST equation of state with the appropriate uncertainty bands. The average start (initial) pressure and associated pressure increase during the compression cycle for each test point is provided in Table 10.

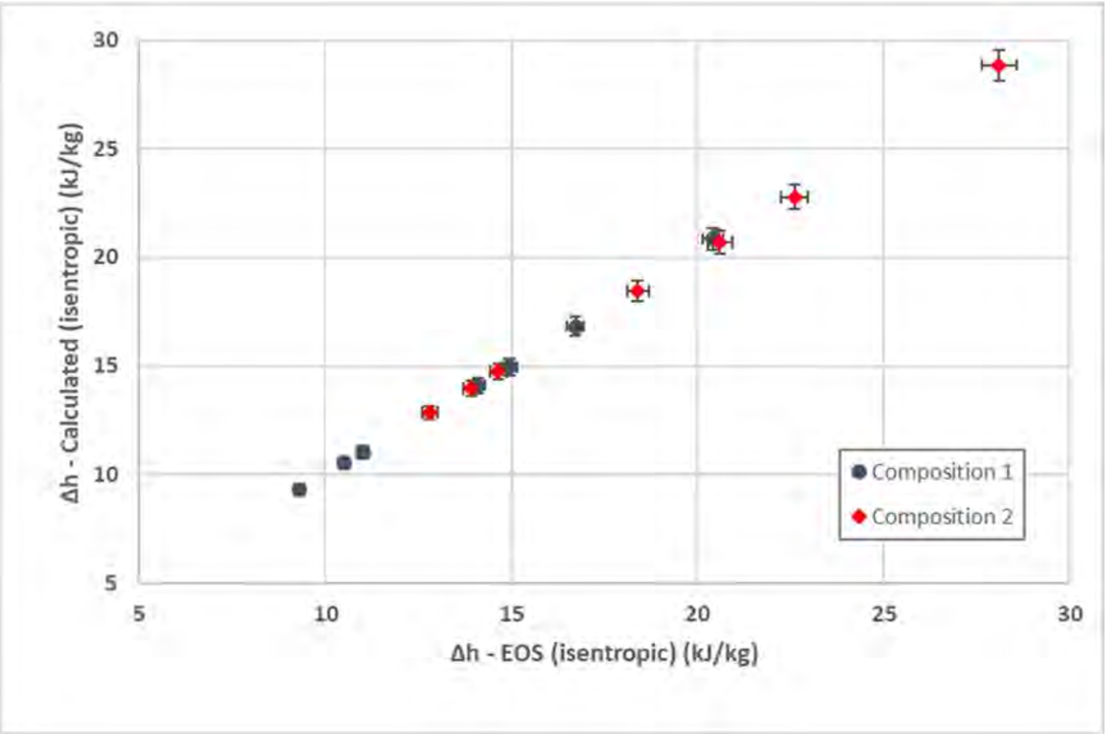


Figure 11. Calculated Enthalpy Rise versus Eos Predicted Values for the Test Gas Compositions

Composition 1		Composition 2	
Start Pressure (P1)	ΔP	Start Pressure (P1)	ΔP
MPa	MPa	MPa	MPa
2.089	0.327	2.075	0.316
3.450	0.543	3.454	0.529
6.220	0.999	6.226	0.985
7.617	1.325	7.610	1.303
10.374	1.928	10.322	1.913
13.099	2.848	13.223	2.945
15.872	4.361	16.027	4.534

Table 10. Average Pressure Rise for each Initial Start Pressure Point

For test points with initial pressures under 13.5 MPa, the difference between the NIST prediction of enthalpy rise and the measured enthalpy rise was below 1% for both gas compositions. For the test points above 13.5 MPa (where the gas phase is in the supercritical region), higher deviations were recorded, where NIST predicted notably lower values for enthalpy rise than what was measured. For the test campaign in this second phase, the pressure differential of the gas varied from 0.32 to 4.53 MPa.

COMPARISON TO EQUATIONS OF STATE

The calculated enthalpy rise of both gas compositions were compared to the default NIST calculations (EOS) found in REFPROP to investigate deviations between measurement and prediction. Measured temperature and pressure conditions were used as input for the enthalpy rise predictions between the two state points from the NIST model. All modeling for the EOS was performed using the NIST REFPROP software version 9.1. Results for the average tested enthalpy rise and average enthalpy rise from the equations of state are provided in Figure 12 and with the associated measurement uncertainty bands.

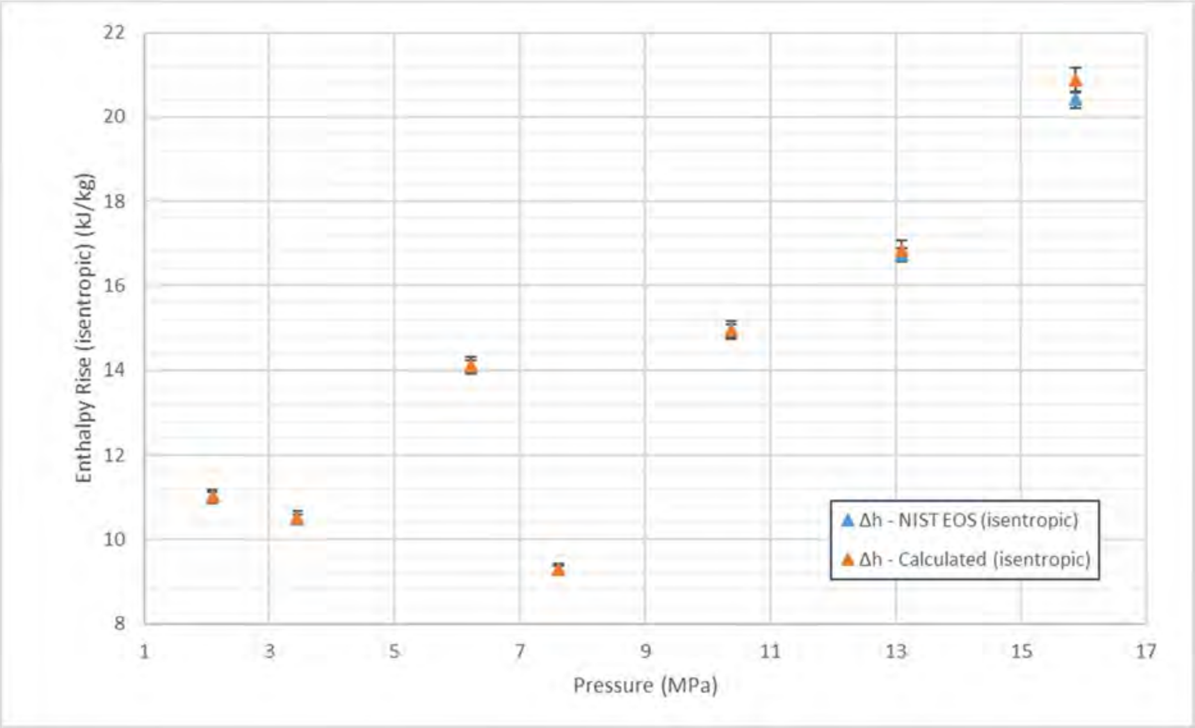


Figure 12. Average Calculated Enthalpy Rise and Average EOS Predicted Enthalpy Rise for Gas Composition 1

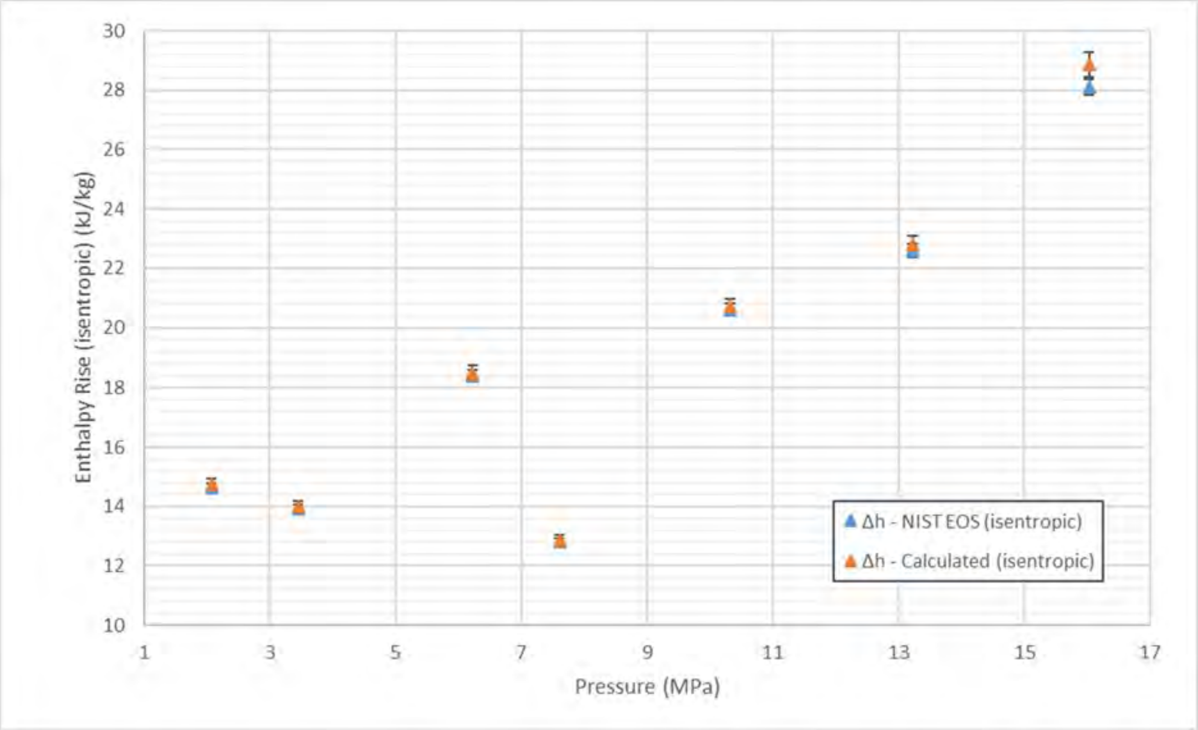


Figure 13. Average Calculated Enthalpy Rise and Average EOS Predicted Enthalpy Rise for Gas Composition 2

For Gas Composition 1, the difference between the measured value of enthalpy rise and the EOS predictions are below 1% for start pressures under 14 MPa, shown in Figure 14. For initial pressure test points under 12 Mpa, the difference is under 0.3%. Within this region, the predicted enthalpy rise from the EOS and measured enthalpy rise are within the associated uncertainty bands. For the 15.87 MPa start pressure test point, however, the EOS provided a difference in enthalpy rise of over 1%, under-predicting what was measured by approximately 2.2%.

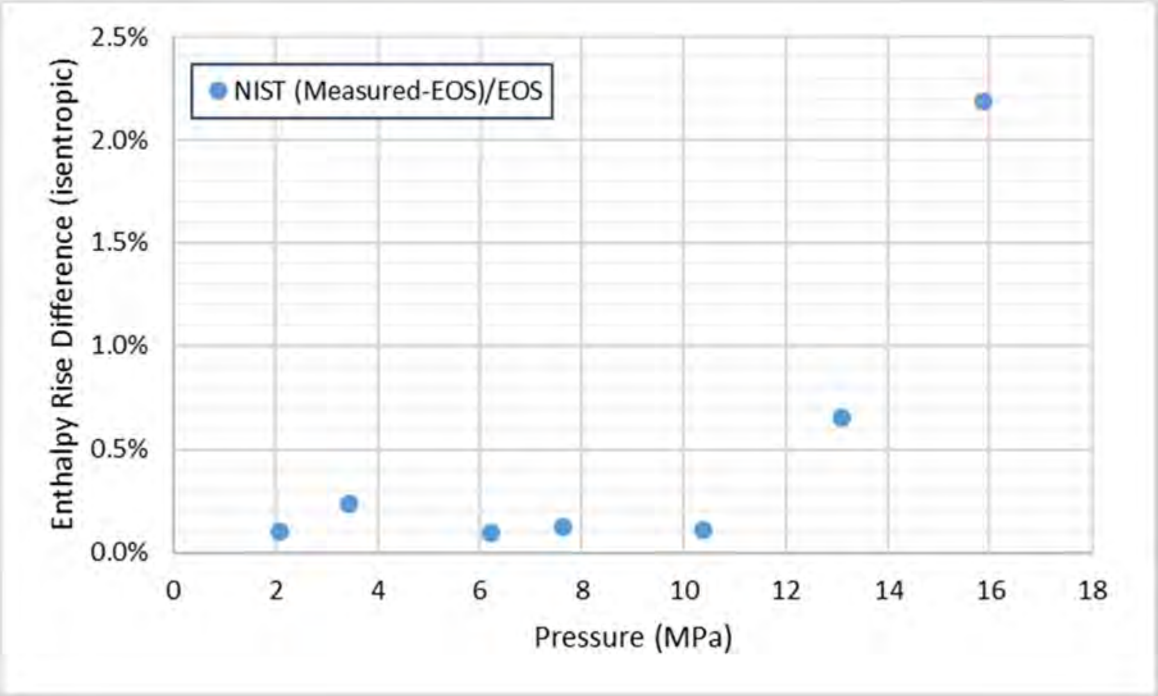


Figure 14. Difference between Measured Enthalpy Rise and NIST EOS Predictions – Composition 1

Similar results are seen for Gas Composition 2. The difference between the measured enthalpy rise and EOS prediction is under 1% for initial pressure test points below 14 MPa, shown in Figure 15. The largest deviation was found at the 16 MPa initial pressure test point, with the EOS under predicting enthalpy rise by approximately 2.7%. Overall, higher deviations between EOS predictions and measured values of enthalpy rise were recorded with Gas Composition 2 than with Gas Composition 1. This may be attributed to the increase in the number of gases in Gas Composition 2 and the inclusion of heavier hydrocarbons that are not present in Gas Composition 1.

For both gas compositions, a trend is apparent that as pressure increases past the critical point of the gas mixture, the EOS prediction of enthalpy rise deviate further from the measured value. It is expected that higher deviations can occur at more elevated pressures than those tested in this project. It is noted that as initial start pressures increased, the pressure differentials that occurred for the same stroke length increased. However, significant changes in the difference between the EOS predictions and measurement of enthalpy rise do not occur until after an initial start pressure 14 MPa.

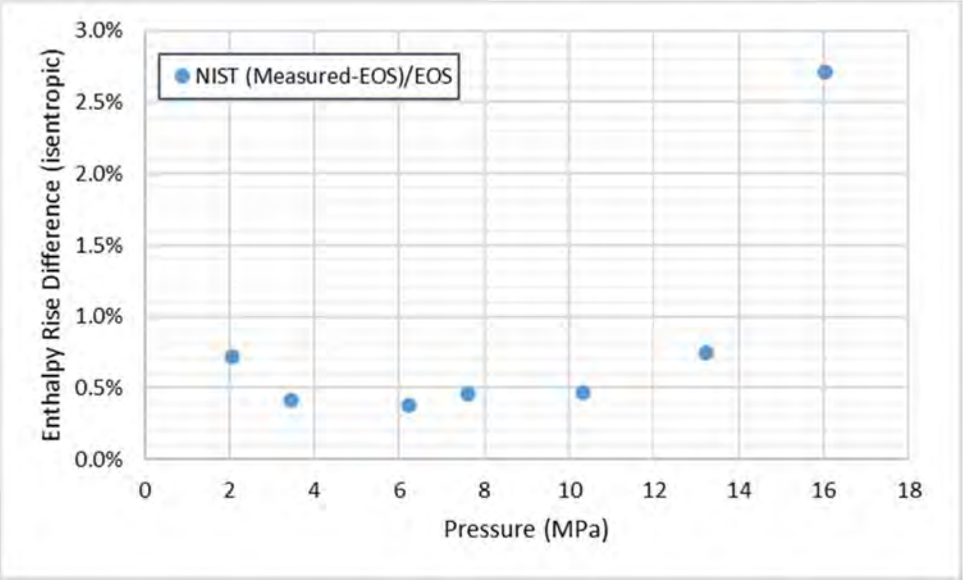


Figure 15. Difference between Measured Enthalpy Rise and NIST EOS Predictions – Composition 2

SUMMARY

Accurate gas property prediction is a necessary component throughout the oil and gas industry for end users, operators and equipment manufacturers for proper sizing and selection of compressors, improving overall efficiency, and reducing operating costs. Equations of state are utilized to predict the thermo-physical gas properties needed for such calculations. However, large uncertainties can be encountered for these properties in the different equations of state that are commonly used, especially when gas compositions differ from typical “clean” transmission grade natural gas. Isentropic enthalpy head (i.e., the enthalpy rise along constant entropy lines) is a critical parameter that is required to accurately design and performance test compressors; therefore, there is significant value in developing a method and test apparatus that can directly measure enthalpy rise for a wide range of gas compositions in the gaseous and supercritical regions of the phase map. A novel method and apparatus was developed for the direct measurement of this property. This technology was previously described by (Brun et al [2018]) and is based on utilizing a calibrated single stroke near-adiabatic compression event to increase the enthalpy of a test gas between two known thermodynamic states.

The test results presented in this paper show that the equation of state enthalpy rise predictions for the two test gases were within the uncertainty bands of the test results for conditions where the mixture was in the gaseous phase (at pressures less than approximately 8.3-9.5 MPa). At pressures where the mixture was in the supercritical region, measured enthalpy began to deviate from the EOS prediction, increasing the further the operating point is from the critical point. This can be accounted for by operators and manufacturers with in-house equation of state solvers (if based on NIST calculations) by adding a appropriate margin to the enthalpy rise for gases in the supercritical region. Higher deviations may be present if in-house solvers utilize other EOS models. Results found in this paper can be compared to various other commonly used EOS model predictions.

Another interesting finding was that the percent difference in the measured versus predicted enthalpy was higher for the gas mixture with significant amounts of heavier hydrocarbons than it was for the gas mixture with equal amounts of methane and CO_2. The mixing rules for gases with more components, particularly ones with heavier hydrocarbons, decrease in accuracy at higher pressures.

These test results are a good beginning in understanding when equation of state predictions start to lose their accuracy for gas mixtures when operating near supercritical areas on the phase diagram or for gas mixtures including atypical components such as heavy hydrocarbons, water or CO_2, similar to those investigated in this test campaign.

NOMENCLATURE

a = Speed of Sound
c_v = Specific Heat
f = Frequency
H = Head
h = Enthalpy
$\triangle h_{isen}$ = Isentropic Head
$\triangle h_{actual}$ = Actual Head
M = Mass
P = Power
p = Pressure
ρ = Density
T = Temperature
V = Volume
v = Specific Volume
$\widetilde{V}$ = Calibration Volume
W = Mass Flow
Z = Compressibility Factor
η = Efficiency

Subscripts

s,1 = Suction
d,2 = Discharge
c = Isentropic
a = Actual
p = Polytropic

FIGURES

APPENDIX

None

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