

# Application of Viper Energy Recovery Expansion Device in Transcritical Carbon Dioxide Refrigeration Cycle

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## Abstract

In light of recent trends towards energy efficiency and environmental consciousness, the heating, ventilation, air conditioning and refrigeration (HVAC&R) industry has been pushing for technological developments to meet both of these needs. As such, several solutions for harnessing the energy released from refrigerants during the free expansion process of a conventional vapor-compression cycle have been developed to increase overall cycle efficiency. Additionally, the implementation of natural refrigerants with reduced Global Warming Potential (GWP) in refrigeration cycles has become increasingly important since current refrigerants are ozone depleting or linked to climate change. This research focuses on the natural refrigerant carbon dioxide (CO<sub>2</sub>) due to its GWP and its significant potential for energy recovery during the expansion process. The goal is to investigate the potential impact of installing an energy recovery expansion device, known as the Viper Expander, into a transcritical CO<sub>2</sub> refrigeration cycle. The Viper Expander operates by using a nozzle to accelerate the high pressure CO<sub>2</sub> into a high velocity jet of fluid impinging on a micro-turbine impeller. The impeller is coupled with a generator, which harvests the kinetic energy of the CO<sub>2</sub> by converting it into electrical energy that can be fed back into one of the system components, such as a fan or compressor motor. The feasibility of this application was analyzed through modeling performance in the unit's proposed test stand for concept validation, and resulted in a model prediction of the Viper harvesting up to 7% of the compressor power consumption at certain operating conditions. Initial experimental testing resulted in a Viper isentropic efficiency of 49.3%. Future work will focus on optimizing Viper design with test conditions, as well as relating Viper performance to overall system performance.

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## 1. Introduction

As the importance of global warming becomes more and more apparent every day, efforts towards developing environmentally friendly solutions are constantly being made. When considering the design of refrigeration systems, several main efforts can be identified. For example, increasing the overall efficiency of the system through addition and optimization of components can be applied in any and all refrigeration applications. Another strategy focuses on the selection of the working fluid used in the system. With the phase out of hydrofluorocarbons (HFC's), which make up a large portion of today's refrigerants, being expedited by legislation all over the world, there is a large need for low-GWP alternative refrigerants. The application of the Viper expansion work recovery device in a transcritical CO<sub>2</sub> refrigeration cycle offers a solution that encompasses both of the efforts described above.

The Viper expander replaces the Thermostatic Expansion Valve (TXV) found in traditional vapor-compression cycles. In such cycles, the work expelled from the system to reduce the refrigerant pressure from condensation pressure to evaporation pressure goes unharnessed. The Viper expander is designed to harness that work by using a nozzle to convert the high-pressure refrigerant flow to a high-velocity stream of refrigerant that impinges on the Viper impeller, causing the impeller to rotate. The impeller is connected to a generator inside the Viper housing, thus converting the kinetic energy of the refrigerant to electrical energy that can be fed back to one of the system's components, such as a fan or the compressor. A CAD mockup of the Viper can be seen in Figure 1.

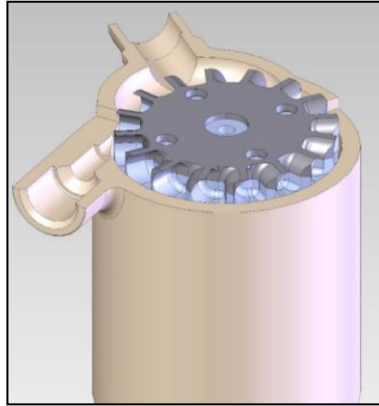


Fig. 1. CAD Mockup of the Viper Expander

Originally, the Viper was developed for an R410A split-system heat pump and tested by Czapla et al. [1]. However, R410A is an HFC with a GWP of 1725. It is envisioned that systems using R410A as the working fluid can be subject to legislative changes in the near future. This increases the attractiveness of  $CO_2$ , as its low GWP reduces the likelihood of installations being impacted by future environmental legislative actions.

In addition to the environmental benefits of utilizing  $CO_2$  as the working fluid, the expansion energy available in a transcritical cycle is significantly increased compared to a subcritical vapor compression cycle. For a cycle to be deemed “transcritical”, its high side pressures must exceed the critical pressure of the working fluid, or 7.4 MPa in the case of  $CO_2$ . Groll and Baek [2] tested a piston-cylinder expander known as the “Expansion Device – With Output Work” and found improvements in the coefficient of performance (COP) ranging from 7.07 to 10.46%. In another study, Yang [3] tested a double-acting rotary vane expander/generator that increased the COP by 11-14%. Both of these findings showed performance increases in the system using fixed-volume ratio expansion devices. Neither study utilized a nozzle and impeller design similar to that found in the Viper expander.

Previous work that utilized an impeller to harness expansion work was that of He [4], where a Pelton-type expansion work recovery device in an R134a refrigeration cycle was tested. The expander in the test by He [4] produced improvements in the system COP and cooling capacity of 6.5% and 5.4%, respectively. While this performance gain is promising, the working fluid was R134a, an HFC with a GWP of 1300. Another strategy for harnessing of expansion work is provided by Elbel and Hrnjak [5]. They tested a prototype ejector in a transcritical  $CO_2$  refrigeration system and the results showed an increase in the COP and the cooling capacity by 7% and 8%, respectively.

To initially assess the feasibility of applying the Viper expander in a transcritical  $CO_2$  refrigeration cycle, a basic transcritical system model was developed based on the work published in Czapla et al. [6]. Next, the physical limitations of the test stand using the operating conditions found during the experiments performed by Hubacher [7] were used to modify the basic system model into a test stand model and to predict more accurate performance result. Once the test stand model was complete, the model was further updated by replacing the expansion valve submodel with a Viper Expander submodel. Finally, the physical test stand was updated and a prototype Viper expander was installed. Initial experimental results were obtained and are reported in this paper.

## 2. Approach

### 2.1 Basic System Model

Czapla et. al. [6] performed a theoretical analysis of the Viper expander performance in a transcritical  $CO_2$  refrigeration cycle. A schematic for the basic system model utilized in this study is shown in Figure 2. In this analysis, the expansion valves were replaced with Viper expanders on both the high-pressure stage (between States 5 and 6) and the low-pressure stage (between States 7 and 8). The potential system performance benefits were calculated and resulted improvements of the COP of the system from 1-11% with the installation of either one or two Viper expanders in the system.

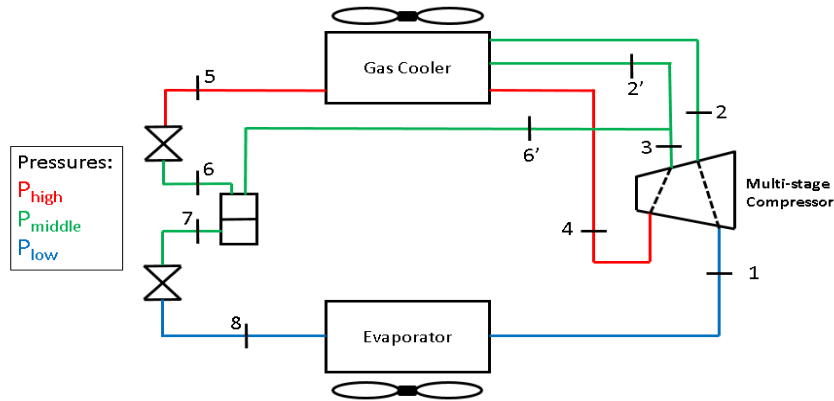


Fig. 2. Basic System Model Schematic, Czapla et. al. [6]

The basic system model utilized the three operating conditions shown in Table 1. These operating conditions were selected because the high-side pressures were above the critical pressure of  $CO_2$  (7.4 MPa).

Table 1. Selected Operating Conditions

Operation	Gas Cooler Temperature [°C]	Evaporator Temperature [°C]	Description
1	38	2	Max gas cooler temp with max evaporator temp
2	38	-7	Max gas cooler temp with min evaporator temp
3	50	-29	Any other operation that should be tested as if both heat pump modes are used

These test conditions were modeled using a computer program called Engineering Equation Solver (EES), and the results from Czapla et. al. [6] were replicated before the basic system model was modified to represent the physical limitations of the test stand discussed in the next section.

## 2.2 Test Stand Model

The test stand being analyzed is a hot gas-bypass compressor load stand. A schematic of the load stand and its measurement points can be seen in Figure 3. The process of the working fluid in this test stand is visualized in a P-h diagram shown in Figure 4.

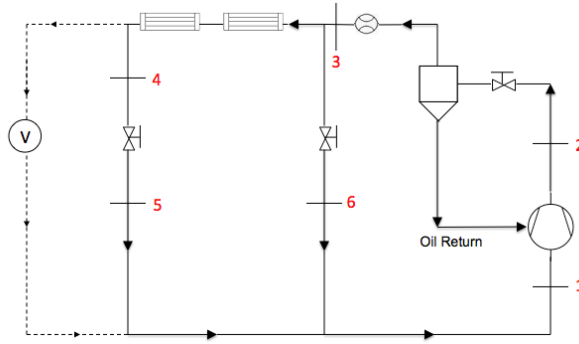


Fig. 3. Hot Gas-Bypass Stand Schematic

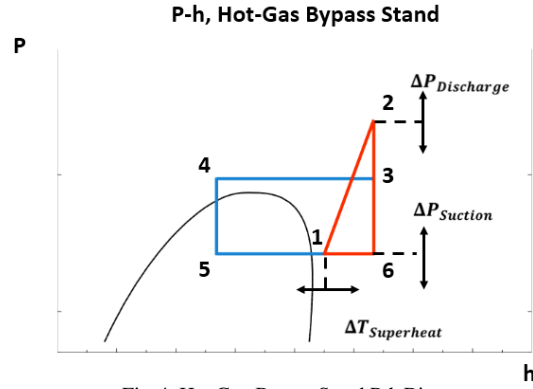


Fig. 4. Hot Gas-Bypass Stand P-h Diagram

The hot gas-bypass begins with the working fluid being compressed from States 1 to 2. After the working fluid is compressed to its outlet at State 2, it is isenthalpically throttled to the intermediate pressure of the system at State 3. After the intermediate pressure is reached, the flow is split. The majority of the flow stays in the hot gas phase and is throttled from the intermediate pressure at State 3 to the suction pressure at State 6. The remainder of the refrigerant continues through the condenser (or gas cooler) and rejects heat to reach State 4. The process from States 4 to 5 is the expansion process where the Viper expander has been installed. This placement is shown in the test stand schematic in Figure 3 by the circular “V” placed in parallel with the conventional expansion valves. At the outlet of expander, this flow is then mixed with the hot-gas flow to return to compressor suction at State Point 1. The return to State Point 1 is achieved through an energy balance shown in Equation 1.

$$\dot{m}_{liquid} * h_5 + \dot{m}_{hotgas} * h_6 = \dot{m}_{total} * h_1 \quad (1)$$

The development of the test stand model started with utilizing experimental data from Hubacher [7]. To include physical limitations of the test stand to be used for Viper testing, test conditions from this experimental data were input into the basic system model, and the model was updated to include heat loss from the compressor as well as pressure drops across the throttling valves, condenser, evaporator, and the Viper assembly.

The compressor heat loss was calculated through a third-order best fit derived as a function of the pressure ratio across the compressor. This equation was developed by Hubacher [7] and is shown in Equation 2.

$$\dot{Q}_{comp} = -3.08511 + 3.12715 * P_r - .641968 * P_r^2 + 0.0489704 * P_r^3 \quad (2)$$

The effect of the heat loss inclusion on compressor performance in the resulting model is illustrated on a P-h diagram in Figure 5. The finalized model for the test stand utilized the code to produce the red “hl” line. The state points of this cycle are included to show the difference created by the inclusion of the heat loss code in the test stand model.

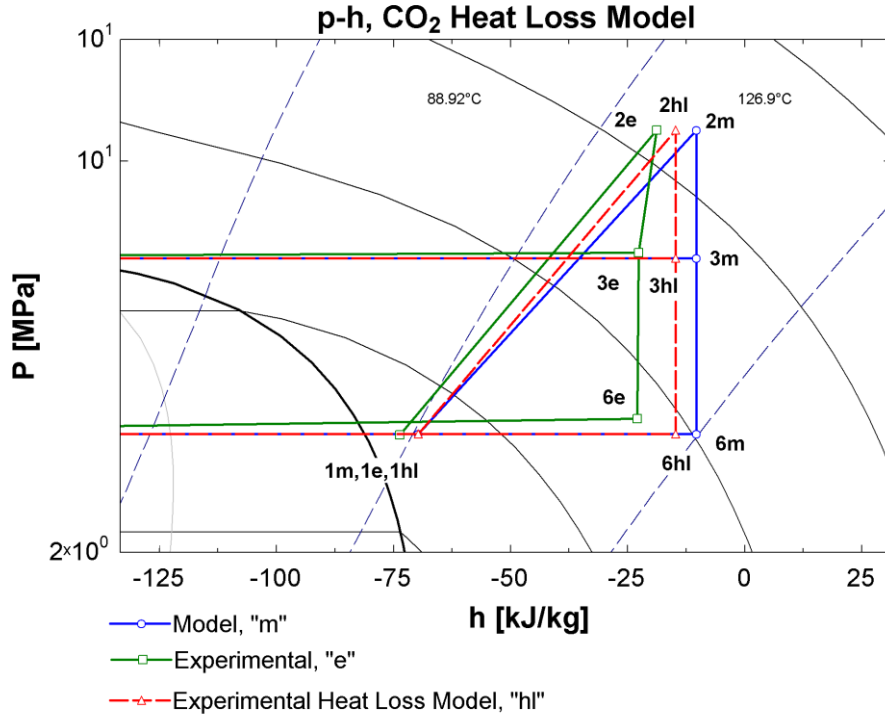


Fig. 5. Compressor Heat Loss Modeling Results

The Viper expander was installed in the hot-gas bypass load stand isolated by a ball valve that allows 0 to 100% of the flow to travel through it. The Viper expander and ball valve were installed in parallel to the conventional expansion valves 1 and 2 to keep the original function of the hot-gas bypass stand. The installation schematic can be seen in Figure 6.

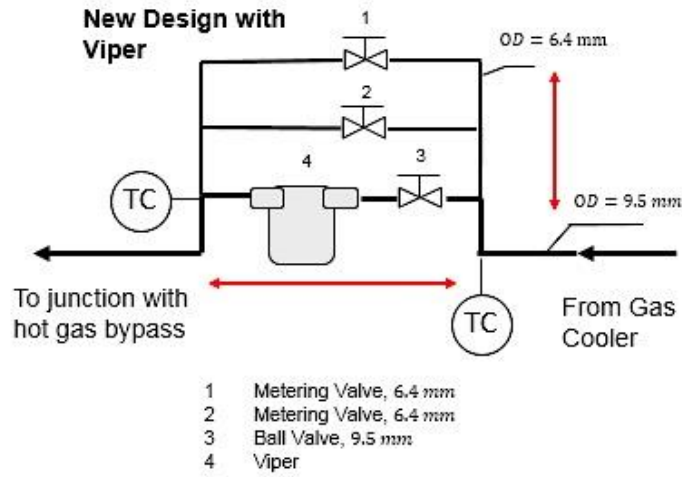


Fig. 6. Inclusion of the Viper in the Liquid Line

While these modifications allow for versatility of the load stand, pressure drops are introduced to the system. To calculate this pressure drop, the governing equation is shown in Equation 3.

$$\Delta P = \lambda * \left( \frac{l}{d_h} \right) * \left( \frac{\rho v^2}{2} \right) \quad (3)$$

Where

$$\lambda = f(Re, k/d_h) \quad (4)$$

Here,  $k$  is the absolute roughness of the surface,  $\nu$  is the fluid kinematic viscosity,  $\lambda$  represents the flow friction coefficient, and density is symbolized by  $\rho$ . The hydraulic diameter,  $d_h$ , is based off of a 9.5 mm OD tube with an ID of 6.4 mm, and is therefore calculated to be 3.1 mm per Equation 5.

$$d_h = 2(r_o - r_i) \quad (5)$$

The length of the tubing,  $l$ , is considered to be a summation of the actual tube length and also the equivalent lengths of the components that the flow passes through. The pressure losses between the Viper expander and the closest upstream and downstream pressure transducers were calculated and tabulated in Table 4 shown in Section 3.2.

### 2.3 Viper Performance Using Test Stand Model

Once the test stand model was developed, the Viper expander performance was analyzed for operation in the liquid expansion line of the hot gas-bypass load stand. Equations 6 and 7 show the calculation of the expander output enthalpy via isentropic efficiency,  $\eta_{is,exp}$ .

$$\eta_{is,exp} = \frac{h_4 - h_5}{h_4 - h_{5s}} \quad (6)$$

Where

$$h_{5s} = f(p_5, s_4) \quad (7)$$

Once the expansion output enthalpy,  $h_5$ , is found, Equation 8 is utilized to find the expansion power output by the Viper expander.

$$\dot{W}_{Viper} = \dot{m}_{liquid} * (h_4 - h_5) \quad (8)$$

The calculation of the power harnessed by the Viper expander makes the term  $\dot{W}_{Viper}$  positive, as  $h_4 > h_5$ . Because the hot gas-bypass load stand has no evaporator, the performance of the Viper relative to the rest of the cycle is calculated as a percentage of the compressor work input, shown below by Equation 9.

$$\% \dot{W}_{Viper} = \frac{\dot{W}_{Viper}}{\dot{W}_{comp}} * 100\% \quad (9)$$

## 3. Results & Discussion

### 3.1 Test Stand Model

The test stand model was able to predict the experimental data. However, despite the accuracy of the expansion process analysis, the approximations of the compressor efficiencies caused several state points to be less accurate. A summary of these values and their percentage errors can be seen in Table 2.

Table 2. Hot-Gas Bypass Comparison of Model and Experimental

Property and State Point	Units	Experimental	Model	% Error
$P_6$	MPa	4.41	4.20	5.03
$P_3$	MPa	7.47	7.35	1.63
$P_2$	MPa	11.03	11.03	0
$T_6$	°C	46.80	55.17	15.17
$T_3$	°C	71.40	78.94	9.55

$T_2$	°C	95.40	100.50	5.06
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The key component of the developed test stand model is the expansion process. Therefore, as long as the liquid mass flow rate through the expansion process, and the pressures and enthalpies at the inlet and outlet of the expander are predicted accurately, the model will accurately predict the expansion work recovery. Sources of future improvements in the model are taking the pressure increase across the superheated portion of the evaporation process into consideration, as well as the pressure drop across the mixing of the hot gas bypass flow with the two-phase working fluid flow.

### 3.2 Viper Performance Model

The predictions of the Viper expander performance using the test stand model yielded promising results. An example of these results and their accuracy relative to the experimental data can be seen in Table 3. It should be noted that these values were calculated with the mass flow rates that were assumed for Operating Condition 1 from the basic system model, not the values calculated when considering the mass flow rate limitations of the test stand. This assumption does not change the percent difference calculations or the pressure drops, simply the power output values. Also, no theoretical values were calculated for the high-pressure stage power output at 30% isentropic efficiency, explaining the N/A shown in Table 3.

Table 3. Comparison of System and Test Stand Model Results

	High Side			Low Side		
	$\Delta P_{exp}$	$\dot{W}_{viper}$		$\Delta P_{exp}$	$\dot{W}_{viper}$	
		$\eta_{is} = 30\%$	$\eta_{is} = 50\%$		$\eta_{is} = 30\%$	$\eta_{is} = 50\%$
	<b>kPa</b>	<b>kW</b>		<b>kPa</b>	<b>kW</b>	
<b>Test Stand</b>	5271	0.1827	0.3045	3661	0.1873	0.3121
<b>System</b>	5481	N/A	0.3133	3871	0.2055	0.3426
<b>Difference (%)</b>	3.83	N/A	2.81	5.42	8.86	8.90

For a more visual representation, two of the same operating conditions were run using the basic system model as well as the test stand model. The two results were overlaid on the same P-h diagram, shown in Figure 7. Because the basic system model data came from a two-stage compressor, the data had to be split into high side and low side iterations. Also, the pressure drops from the discharge valve and gas cooler in both the high and low pressure stage cycles are included in the P-h diagram. In Figure 7, the green line represents the basic system cycle whereas the blue and red dotted lines represent the low and high stage test stand cycle, respectively.

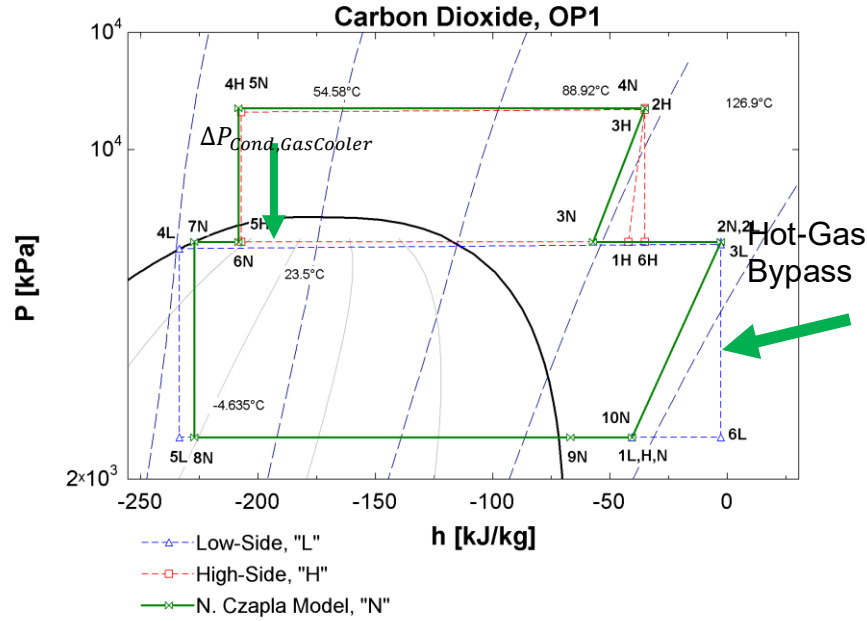


Fig. 7. Hot Gas-Bypass Stand, Experimental Versus Test Stand Model P-h Diagram

Several key attributes of the above comparison are worth noting. First, States 6N and 3N should be connected by a line representing the transfer of saturated vapor from the flash tank in the basic system model. This was left out in order to offer easier visualization of the effects of the pressure drop across the condenser and discharge valve. This pressure drop shown between the heat absorption between States 5H and 1H and the condensing between States 3L and 4L makes the pressure drop in the latter scenario more apparent, and is highlighted with a green arrow. Also, the fundamental difference in the hot gas-bypass stand principles and a standard transcritical cycle is clearly displayed. Note the right triangle created from the isenthalpic expansion from the compressor output to State 6 in the hot gas-bypass cycles and its absence in the green, basic system model line. This difference is due to the inclusion of an evaporator in the basic system cycle, which provides the enthalpy increase to reach the suction state. This is in contrast to the hot-gas bypass stand needing a higher-enthalpy flow to create an energy balance in order to reach the compressor suction state.

In order to increase the test stand model's accuracy, the effects of the pressure drop across the test stand upstream and downstream of the Viper have been calculated and are shown below in Table 4. These calculations were introduced and discussed in Section 2.2 of this paper.

Table 4. Pressure Drop Results

Operation	$\Delta P_{Upstream}$ [kPa]	$\Delta P_{Downstream}$ [kPa]	Work Loss [% $\dot{W}_{Viper}$ ]
1	161.1	40.27	3.10
2	55.72	13.93	0.32
3	97.49	24.37	0.54

These results show that the power lost from the Viper expander is proportional to the pressure drop throughout the tubing, which is to be expected. However, pressure loss is not the only factor that has an effect on this loss of work. The temperatures at the output of the gas cooler are also relevant, as the entropy and enthalpies used in this calculation are dependent upon those values.

Sources of future improvement of this model are similar to those discussed in Section 3.1, as any shortcomings of the test stand model affect these results inherently. However, to further improve the accuracy of the values of estimated power output from the Viper expander, the pressure drop from the hot gas line pressure transducer to the mixture point of the hot gas and liquid lines should be calculated and considered in order to provide the most accurate Viper expander output pressure.

The largest limiting factor of the Viper expander performance under the three transcritical operating conditions discussed in Section 2.1 was the experimental mass flow rate of the stand. The mass flow rate is limited to 0.167 kg/s, and was taken into consideration when comparing the results of the basic system model to the test stand model. With an assumed isentropic efficiency of 50%, Table 5 shows the final model results for the potential work recovered by the Viper in the test stand model for the three transcritical operations.

Table 5. Results of the Viper Power Recovery

Operation	$\dot{m}_{liquid}$ [kg/s]	$\dot{W}_{viper}$ [kW]	% $\dot{W}_{Compressor}$
1	0.0215	0.286	3.30
2	0.0254	0.155	2.06
3	0.0385	0.313	3.71

As was discussed earlier, the power recovered by the Viper is directly proportional to the mass flow rate through the expansion process. Another main factor is the change in enthalpies, which are functions of pressure, temperature, and entropy in the case of isentropic efficiency calculations. The mass flow rate through the liquid line,  $\dot{m}_{liquid}$ , was calculated through the energy balance about the compressor suction point discussed in the analysis section of this paper. The values shown here are considerably less than the total mass flow rate through the compressor, and the reasoning for that goes back to the fundamentals behind the separations of flow in the hot gas-bypass stand.

### 3.3 Experimental Viper Performance

When the Viper expander was tested in the hot-gas bypass stand, an isentropic efficiency of 49.3% was obtained. This result was found while the stand was not able to reach optimal or modeled testing conditions due to an underpowered compressor. In particular, the test stand was unable to reach sub-cooling at State Point 4 and produce any significant pressure rise. Therefore, data that had proven superheat at the compressor inlet and a water side energy balance in the condenser was utilized to find the Viper inlet and outlet state points and determine the isentropic efficiency. It was calculated that the inlet quality to the Viper was 46.7%, producing the 49.3% isentropic efficiency of the Viper stated above. The resulting P-h from the experimental data can be seen below in Figure 8.

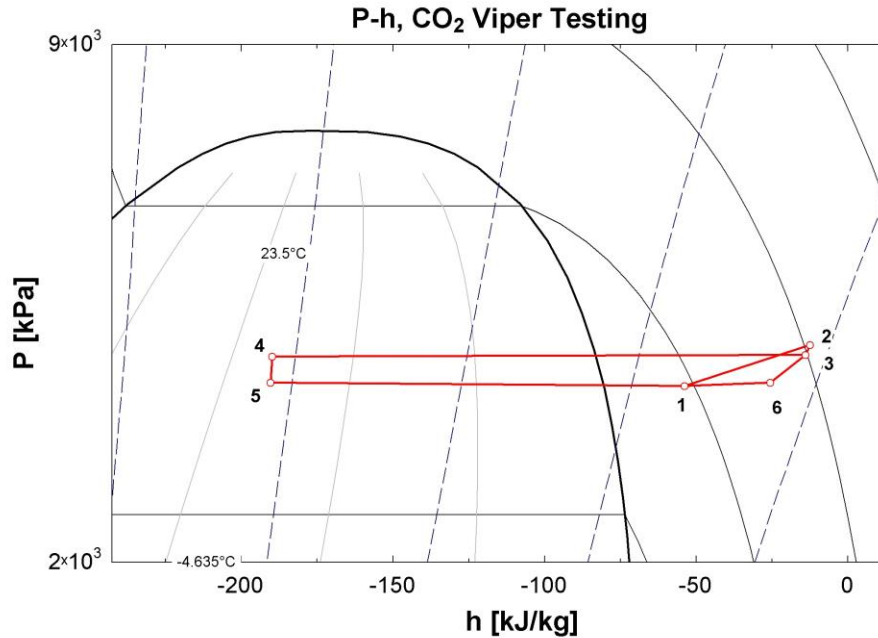


Fig. 8. Logarithmic P-h Diagram of Viper Experimental Results

A more favorable result would come from a cycle where there was adequate superheat and sub-cooling, as well as a reasonably large difference between intermediate and suction pressures. This would allow the enthalpies in

the energy balance to be constant, eliminating the calculation of heat transfer in the condenser, and also provide more potential energy to harness through the expansion recovery. Regardless, these results validate the concept, and provide motivation to continue to develop and optimize both the test stand and Viper expander design.

#### 4. Conclusions & Recommendations

This paper presented a theoretical analysis of the performance of a Viper expander in a transcritical  $CO_2$  hot-gas bypass stand. Inherent physical losses of the test stand were included in the model and validated with experimental data, thus proving the ability of the model to accurately represent real test stand performance. The results of the preliminary testing of the Viper also offered concept validation of the Viper expander, even though the model test conditions could not be replicated due to poor compressor performance. The Viper expander performed at an isentropic efficiency of 49.3%. Given the less than ideal test conditions, this is a promising result, which proves that the Viper was capable of replacing a TXV and harnessing expansion work in a transcritical  $CO_2$  cycle.

Recommendations for improvement are to power the stand with an adequate compressor and balance the charge. The charge needs to be fine-tuned in such a way that adequate superheat and sub-cooling can be maintained while still sending a high percentage of the  $CO_2$  through the liquid line and, in turn, the Viper expander. Future work will include experimentally reaching modeled test stand conditions for comparison of theoretical and experimental results, as well as optimization of the Viper expander design to increase its performance.

#### Nomenclature

A	Area	$m^2$	<b>Acronyms</b>	
h	Specific enthalpy	$kJ/kg$	COP	Coefficient of Performance
k	Absolute roughness	$m$	GWP	Global Warming Potential
L	Length	$m$	HFC	Hydrofluorocarbon
$\dot{m}$	Mass flow rate	$kg/s$	TXV	Thermostatic Expansion Valve
p	Pressure	$kPa$	ID	Tubing Inner Diameter
$\dot{Q}$	Heat transfer rate	$kW$	OD	Tubing Outer Diameter
r	Radius	$m$		
Re	Reynolds Number	—	<b>Subscript</b>	
s	Specific entropy	$kJ/kg - K$	1,2,3...	State points
T	Temperature	$^{\circ}C$	comp	Compressor
u	Velocity	$m/s$	exp	Expander
$\dot{W}$	Power	$kW$	hotgas	Hot gas
<b>Greek Symbols</b>			i	Inner
			in	Inlet
			is	Isentropic
			liq	Liquid
			o	Outer
			out	Outlet
			Viper	Viper
$\eta$	Efficiency	$kg/m^3$		
$\rho$	Density			
$\lambda$	Flow friction coefficient			
$\nu$	Kinematic viscosity	$m^2/sec$		

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