Theoretical Analysis of the Impact of an Energy Recovery Expansion Device in a CO2 Refrigeration System

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Theoretical Analysis of the Impact of an Energy Recovery Expansion Device in a CO_2 Refrigeration System

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ABSTRACT

Carbon dioxide (CO_2) is being widely used as a refrigerant in HVAC&R applications due to its low Global Warming Potential (GWP). There are many aspects of CO_2 systems that make it unique to other traditional refrigerants in that it has higher pressure levels and typically operates at transcritical levels. Up to 40% of the losses in a transcritical CO_2 cycle can be due to throttling across the expansion valve. These significant losses make CO_2 systems ideal for installing an energy recovery expansion device that consists of a nozzle, micro-turbine and a generator. The expander has been developed by Regal Beloit Corporation and will be referred to as the Viper Expander. The device functions by using a nozzle to convert the high pressure of the refrigerant into a high velocity jet that is directed into the impeller of the micro-turbine. The turbine impeller then spins a shaft that is coupled with a generator to generator electrical energy. The Viper Expander is to replace the standard passive expansion device. Experimental testing of this device with R410A indicates that the device performs better for refrigerant with low viscosity and high pressure. For these reasons, the implementation of this energy recovery device in a CO_2 refrigeration system has been investigated. The results of this paper quantify the potential impact that this device could have in the system by predicting theoretical recoverable power and improvements to the coefficient of performance (COP). The analysis predicts that the Viper Expander can improve the COP from 1 - 11% and indicates that further experimental investigation should be conducted to verify the results.

1. INTRODUCTION

In 1987 the Montreal Protocol ruled to begin phasing out ozone depleting substances like chlorofluorocarbons (CFCs). The results of this international agreement led to the discovery of non-ozone depleting hydrofluorocarbons (HFCs) such as R134a and R410A. Even though HFCs are not contributors to ozone depletion, they do have significant global warming potentials. As the need to reduce the impact on climate change grows, efficient systems using alternative refrigerants must be developed. Since the natural refrigerant carbon dioxide (R744) is currently being advocated to replace CFCs and HFCs, there has been a renewed interest to develop more efficient CO_2-based system. The reason for this is that CO_2 has many advantages when compared to standard refrigerants and system components are now able to be manufactured to properly operate with CO_2.

Carbon dioxide has a global warming potential (GWP) of 1 and acts as the basis of comparison for all other substances. It is a non-toxic, colorless and odorless substance that is the by-product of many industrial applications. Therefore as significant amounts of CO_2 are being produced, it is a more cost-effective solution than traditionally manufactured refrigerants. Additionally, CO_2 also has many unique thermophysical properties that make it favorable as a natural refrigerant. These include a high heat transfer coefficient, very low viscosity as well as having low pressure drop in piping and heat exchangers. It has also been shown that CO_2 has a lower compression ratio of 20-50% less than HFCs and ammonia, which leads to increased volumetric efficiencies. The operating pressures of CO_2 are also much higher than traditional refrigerants and its critical point is very close to normal operating conditions. Therefore such systems are typically run transcritically and have different designs than pure subcritical systems. The system components must be designed and constructed out of materials to handle the high pressures.

Current commercial designs of CO_2 systems include transcritical and subcritical or cascade systems. One of the main applications for transcritical CO_2 systems is in commercial refrigeration for supermarkets. In the United States, roughly 40% of the electricity consumption is due to refrigeration (Energy Star, 2008). Therefore, there is a huge potential for savings if energy efficient CO_2 systems are designed. A year-long study funded by the U.S. Department of Energy in 2015 showed that a transcritical CO_2 system consumed 6-21% less electricity than an HFC unit during the winter and spring months. A milestone was reached in January of 2014 when The Coca-Cola Company announced the installation of its 1 millionth natural refrigerant CO_2 cooler. The result of this venture for more sustainable refrigeration has resulted in preventing the emission of 5.25 million metric tons of CO_2 into the Earth's atmosphere. By switching to CO_2 as the working fluid for coolers has assisted in allowing The Coca-Company to improve the energy efficiency of its systems by 40% (The Coca-Cola Company, 2014).

Aside from these applications, numerous other uses for CO_2 as a natural refrigerant have been found. As stricter energy regulations and emissions standards are being imposed on the automotive industry, many automotive manufactures are focusing on designing CO_2 based air-conditioning systems for their vehicles. The U.S. Army has invested in research of military environmental control units (ECUs) with CO_2. Li and Groll (2004), discovered that CO_2 based ECUs demonstrate both a higher cooling COP and a higher cooling capacity as compared to traditional R22-based ECUs.

Due to the potential for an energy recovery, a considerable amount of research has been conducted. For example, Yang et al., (2009) performed an experimental investigation of installing a double acting rotary vane expander into a transcritical CO_2 cycle. An electrical generator was connected to the shaft in order to consume the recovered power. Tests were conducted at expansion inlet pressures ranging from 7.85 – 8.35 MPa and the power output from the expander ranged from 348 – 379 W. This corresponded to an increase in COP of 11.1 – 14.2%. For this expander the isentropic efficiency was recorded to be between 19.1 – 22.63%.

Another work output expansion device for transcritical CO_2 systems was developed by Baek and Groll (2005). The device was chosen to be a piston cylinder expander and is known as the Expansion Device-With Output Work (ED-WOW). Three different conditions were tested with indoor and outdoor room temperatures of 20°C and 35°C, respectively The increase in COP ranged from 7.07% to 10.46% which includes the increased cooling capacity resulting from the non-isenthalpic expansion process.

2. THE TRANSCRITICAL CO_2 SYSTEM

The fundamentals of the transcritical system CO_2 refrigeration system investigated will now be presented. The system is comprised of an evaporator, multi-stage compressor, gas cooler with intercooling, two TXVs as well as a flash tank. The schematic of this cycle can be seen below in Figure 1 (left) and the respective log p-h diagram in Figure 1 (right). Detailed specifications of the system's components have been held confidential at the request of the manufacturer.

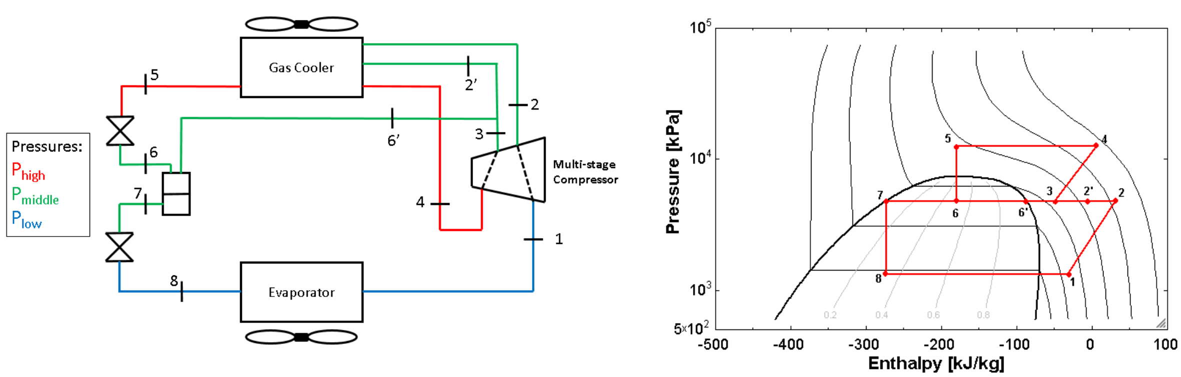

Figure 1: Schematic of transcritical CO2 cycle (left) and log p – h diagram (right). State points: 1- 1st stage suction, 2- 1st stage discharge, 2'- Intercooler exit, 3- 2nd stage suction, 4- 2nd stage discharge, 5- Gas cooler exit, 6- 2nd expansion exit, 6'- Separator exit (sat. vapor), 7- Separator exit (sat. liquid), 8- 1st stage expansion outlet/evaporator inlet

The CO_2 starts by entering the multi-stage compressor at State Point 1 and is compressed to the middle system pressure at State Point 2. Then the refrigerant enters the intercooling portion of the gas cooler which reduces the temperature to State Point 2'. The superheated vapor at State Point 2' is then mixed with saturated vapor at the same middle pressure. This mixing further reduces the temperature of the refrigerant that is to enter the 2nd stage of the compressor. This process is important because it reduces the discharge temperature from the 2nd stage of the compressor. The temperature at state point 4 cannot exceed certain values because the system may not be rated to handle such high temperatures. From State Point 4 to State Point 5 the CO_2 exchanges heat with the surroundings in the gas cooler at a transcritical state. This is ideally a constant pressure heat exchange process. State Point 5 is the inlet to the high side expansion process. The standard unit uses a TXV and isenthalpically expands from State Point 5 to 6. The liquid and gaseous CO_2 is then separated at State Point 6. The saturated vapor is considered to be 6' and the saturated liquid is 7. The saturated vapor is mixed with State Point 2' to determine State Point 3 and the saturated liquid at 7 goes through the low side expansion process to State Point 8. The cycle is then completed as the refrigerant exchanges heat with the ambient air in the evaporator and is heated to the suction temperature at State Point 1.

2.1 Analysis of Transcritical CO_2 System

The analysis will be performed over a range of operating conditions. Five different operating conditions are chosen for the investigation. These five operating points offer a wide range of system performance which will allow for the Viper's potential impact to be thoroughly investigated. The high and low side temperatures of each chosen operating conditions are shown in Table 1.

Table 1: Operating points for analysis

Operating Point (OP)	Low-Side Temperature		High-Side Temperature	
	[°F]	[°C]	[°F]	[°C]
1	-22	-30.0	77	25.0
2	30	-1.11	77	25.0
3	32	0.00	100	37.8
4	-20	-28.9	100	37.8
5	-22	-30.0	122	50.0

The baseline system data for all operating conditions is shown in Table 2. The data in this table was used to find the main system state points that are necessary to construct the cycle in a log p-h diagram. Both temperature and pressure have been experimentally measured for each of the five system operating conditions. Only three pressures are reported: high, middle and low. The pressure difference between all the state points of a particular pressure level is considered to be negligible. Additionally the total system power consumption is reported. These values will be used when determining the COP.

In order to fully construct the cycle, the inlet to the 2nd stage of the compressor (State Point 3) must be calculated. Applying an energy balance to the mixing process at the 2nd state compressor inlet will enable State Point 3 to be determined. Refrigerant from the gas cooler exit (State Point 2') is mixed with saturated vapor from the flash tank (State Point 6'). This mixture then enters the compressors as the 2nd stage suction (State Point 3). The process is shown in the log p–h diagram in Figure 2.

Table 2: Set of given data for all operating conditions

	Operating Point	Units	OP1	OP2	OP3	OP4	OP5
	Test Condition Temperatures	C	-30.0/25.0	-1.11/25.0	0.0/37.8	-28.9/37.8	-30.0/50.0
Press	High-Side Discharge	kPa	7496.3	7717.9	12071.6	9989.1	12398.8
	Middle	kPa	3568.9	4405.5	6590.0	4194.4	4845.1
	Low-Side Suction	kPa	1170.2	2978.0	2718.4	1115.7	1068.5
	Total Power	W	3256.6	2730.7	9254.8	8431.2	9326.2

The first law of thermodynamics states that energy must be conserved for a thermodynamic system. By assuming steady state, adiabatic process, no work and neglecting potential and kinetic energy change, the first law of thermodynamics for the mixing process at State Point 3 reduces to:

$$\dot{m}_{6'}h_{6'} + \dot{m}_{2'}h_{2'} = \dot{m}_3 h_3 \tag{1}$$

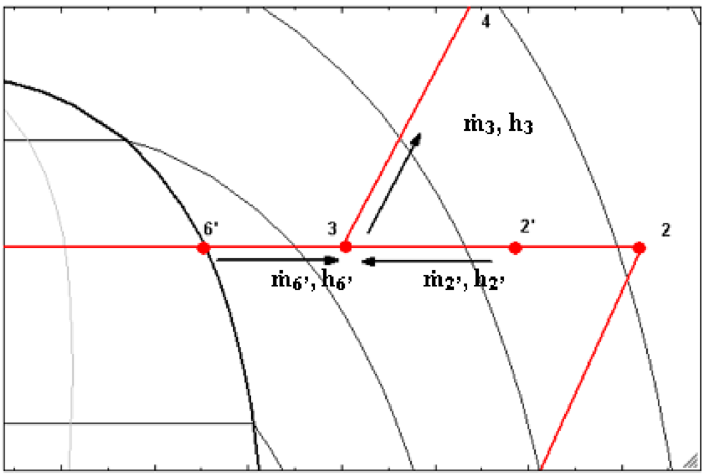

Figure 2: Energy balance for determining State Point 3

Therefore the enthalpy at the inlet to the 2nd stage of the compressor can be solved and the complete cycle can now be determined. In order to understand the potential impact of an energy recovery expansion device, a 2nd law analysis needs to be performed on both the high and low side expansion processes. The maximum available power can be predicted by assuming the expansion processes occur isentropically. This means the inlet entropy will be equal to the outlet entropy.

$$s_{out,is} = s_{in} \tag{2}$$

The outlet isentropic enthalpy is then calculated with the outlet pressure and the outlet entropy:

$$h_{out,s} = f\left(p_{out}, s_{out,is}\right) \tag{3}$$

The total available power from the throttling process can now be calculated as the product of the mass flow rate and difference in enthalpy values for the isentropic process:

$$\dot{W}_{is} = \dot{m}\left(h_{out,is} - h_{in}\right) \tag{4}$$

3. THE VIPER EXPANDER

The Viper Expander is a power generating device that consists of a nozzle, turbine and a generator. The purpose of the expander is to harvest the normally wasted work potential during the expansion process of heat pumps or refrigeration systems. The device is to replace the system's thermostatic expansion valve (TXV) or electronic expansion valve (EXV). The nozzle is used to convert the high pressure energy of the refrigerant at the outlet of the condenser into kinetic energy in the form of low pressure, high velocity jet. The refrigerant will impinge on the impeller causing it to spin and the generator to produce a 3-phase alternating current (AC). The current is then to be rectified to direct current (DC) and fed to an electronically commutated motor (ECM). ECMs operate with direct current and have many advantages over standard AC motors like higher efficiency and lower cost. The power from the Viper Expander will be used to partially power the fan and thus reduce the total system power consumption. Additionally, the work extracted from the Viper Expander will result in a lower outlet enthalpy than the inlet to the nozzle. This reduction in enthalpy is due to the expansion process now being pushed towards partially isentropic which causes an increase in the cooling capacity. Both the reduced power consumption and added cooling capacity should increase the system COP.

The particular Viper Expander design to be tested with the transcritical CO_2 system is based off of an impulse turbine design. For pure impulse turbines, all of the pressure energy is converted into kinetic energy across the nozzle, which correlates to the refrigerant exit velocity. Since the power corresponds to the kinetic energy of the refrigerant impinging on the impeller, the goal is to ensure that as much pressure drop as possible will occur across the Viper nozzle. Typical designs for such impulse impellers are Pelton wheels which have two symmetric buckets with a splitter blade in the center. The splitter blades act to balance the fluid forces acting on the impeller and the buckets cause the fluid flow direction to turn. This magnitude of the change in flow direction relates to the rotational speed of the impeller. The proposed expander design is shown in Figure 3. The nozzle is on the left, the housing is in the middle and the impeller is on the right. The nozzle is a separate piece and screws into the housing.

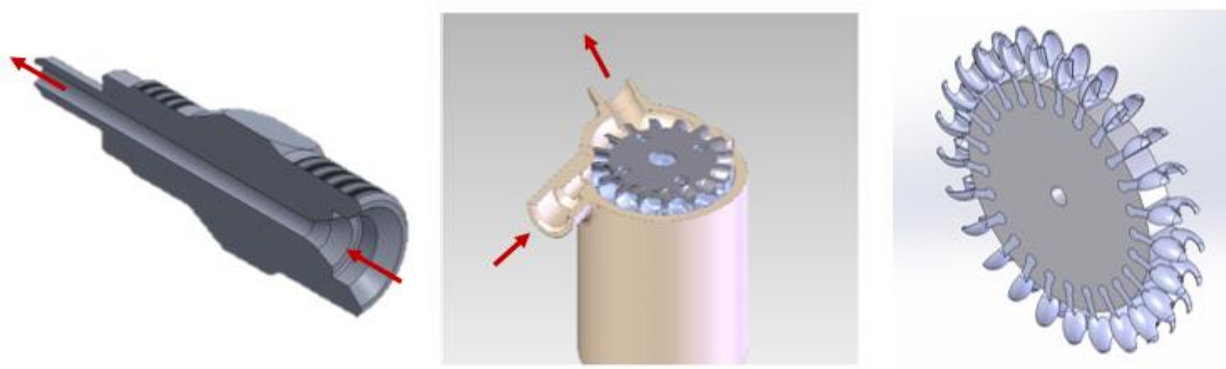

Figure 3: The nozzle (left), housing (middle) and impeller (right) for the expander

The nozzle shown in Figure 3 (left) has a very short converging portion followed by a long constant diameter cross-sectional area. Since this is a two-phase expansion process, the flow will continue to accelerate through the constant diameter section due to the decreasing density as the fluid expands into the two-phase dome. The velocity of the refrigerant can be calculated as follows:

$$V = \frac{\dot{m}}{\rho A} \tag{5}$$

The power generated from the Viper Expander can then be determined by the following equation:

$$\dot{W}_{Viper} = \eta_{Viper}\left(\frac{1}{2}\dot{m}V^2\right) \tag{6}$$

Additionally, the power generated from the Viper Expander can also be determined from the enthalpy change across the expansion process:

$$\dot{W}_{Viper} = \dot{m}\left(h_{out} - h_{in}\right) \tag{7}$$

Since this is a transcritical system with two expansion valves, there are three different possibilities to install the expander in the system. The first possibility is to install the device on the high-side, the second possibility is to purely install it on the low-side and the third possibility is to install it on both the high and low-side. Figure 4 shows the expander installation options in more detail.

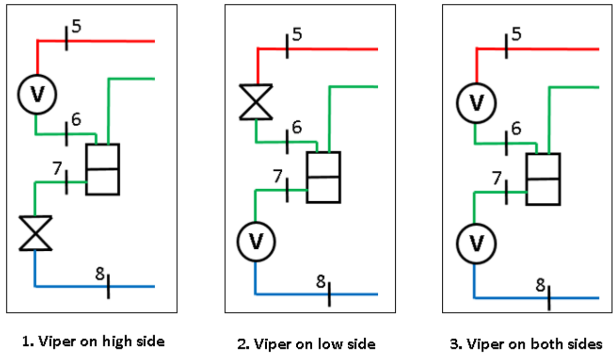

Figure 4: The three possibilities for installing the expander

4. RESULTS

The results of implementing each of the three Viper Expander installation options with all five of the operating conditions were analyzed. Before presenting the impact of the expander on the cycle, it is important to understand how much power is actually available to be harvested. Figure 5 shows isenthalpic and isentropic expansion processes for both the high and low side expansion.

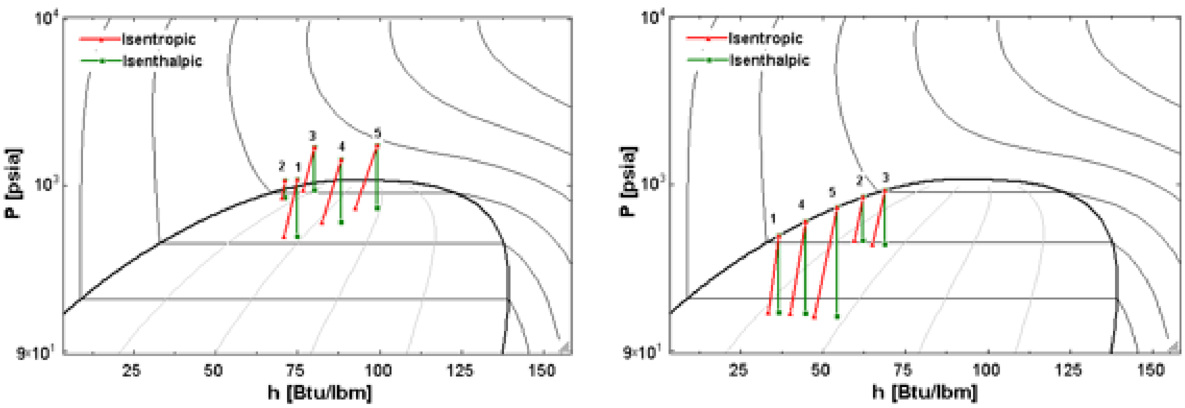

Figure 5: High expansion process (left) and low expansion process (right)

The estimated power to be recovered from the Viper Expander can be found by multiplying the available power during the isentropic expansion process by the target efficiency of the device. The power output can be determined as follows:

$$\dot{W}_{Viper} = \eta_{Viper} \dot{W}_{is} \tag{8}$$

Based on prior experimental data, the maximum achievable value of the expander efficiency has been predicted to range from 30 – 50%. For the sake understanding the theoretical power to be harvested, both the high and low side expander efficiencies were assumed to be 50%. These are just estimates to gather a general idea of how the Viper Expander may perform in the actual CO_2 system. Table 4 shows the available power for each operating point and how much power can potentially be recovered if the aforementioned efficiencies are achieved.

Table 1: Comparison between available power and potential recovered power by Viper Expander

Side	Power	Unit	OP1	OP2	OP3	OP4	OP5
High Side	Available	W	182.5	43.2	606.6	563.2	859.2
	Viper η = 50%	W	91.2	21.6	303.3	281.6	429.6
Low Side	Available	W	85.2	86.1	530.5	241.7	356.3
	Viper η = 50%	W	42.6	43.0	265.2	120.8	178.2
	Viper η = 30%		25.6	25.8	159.2	72.5	106.9

Figure 6 shows the high and low-side expansion processes for standard isenthalpic process, the isentropic process as well as the Viper process. This processes depicts the values of OP1.

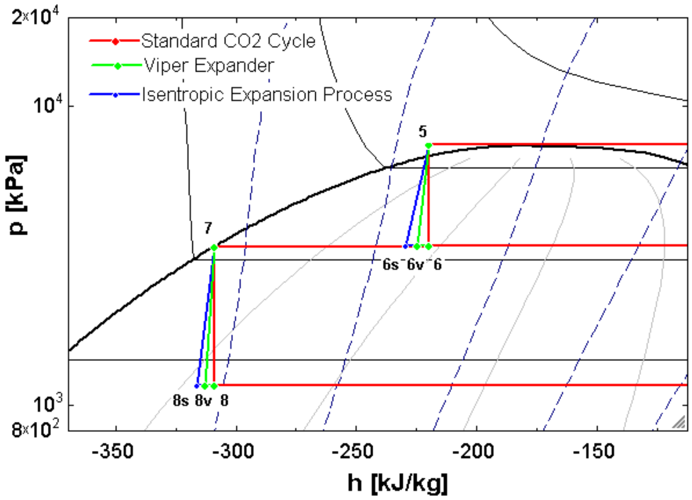

Figure 6: Isenthalpic, isentropic and Viper expansion processes

Observing the power output from the different operating conditions, it is seen that the operating condition with the largest potential power to be recovered is OP5 (50.0°C/-28.9°C) followed by OP3 (37.8°C/0.0°C) and then OP4 (37.8°F/-28.9°F). The available power for these operating points ranges from 563 to 859 W, which correlates to 6 to 9% of the total system power consumption. Also in comparing the high and the low side expansion processes, it is seen that the most power can be generated from the high side expansion in 4 of the 5 cases with the exception of OP2 (25°C/-1.1°C). The reason for this is because of the very low pressure ratio on the high side. Aside from observing the potential power that can be recovered, the impact on the COP will also be compared for the standard system and the system with the Viper Expander. The refrigerant-side COP can be calculated with the following equation:

$$COP_{ref} = \frac{\dot{Q}_C}{\dot{W}_{net}} \tag{9}$$

Table 5 displays the impact on COP for all operating points, Viper efficiencies and Viper installation options. The percentage improvement to the COP, ξ, is also displayed and has been calculated as:

$$\xi=\left(\frac{COP_{Viper}}{COP_{ref}}\text{-}1\right)\times 100\% \tag{10}$$

Table 2: Viper COP Improvement in CO_2 with Viper Efficiencies of 50% and 50%.

OP	Baseline	Viper Installed on:					
	COP	High Side		Low Side		Both Sides	
		COP	ξ	COP	ξ	COP	ξ
1	0.955	0.977	2.30	0.983	2.96	1.006	5.36
2	1.198	1.210	1.00	1.235	3.07	1.247	4.11
3	1.249	1.293	3.50	1.336	6.92	1.384	10.81
4	0.608	0.643	3.23	0.633	4.07	0.653	7.48
5	0.526	0.552	5.02	0.555	5.46	0.583	10.86

Upon investigating the predicated improvement in reported in Tables 5, it is shown that there is a significant potential for the Viper Expander in this transcritical CO_2 system. The impact of the Viper being installed on just the high side could be as high as 5% improvement and the Viper being installed on just the low side could range from 3 to 7% improvement. Additionally, the impact of installing two Vipers into the system would offer the largest improvement. These predicted improvement values range from 4 to11% improvement in COP.

5. CONCLUSIONS AND FUTURE WORK

This paper presented a theoretical analysis of installing an energy recovery expansion device into a transcritical CO_2 refrigeration system. The Viper Expander is a device that consists of a nozzle, impeller and a generator which connects to an ECM of a fan to reduce the power consumption. The device is to replace either the TXV or the EXV of the system. Experimental baseline data from 5 different operating conditions were used to investigate the potential power available and to estimate the impact that the Viper Expander could have in the system. The results show that the system COP can be improved from 1-11% by installing one or two Vipers into the system. The predicted results indicate that further experimental research with the device is worth pursuing.

The future work of this project is to manufacture the described Viper Expander and test it at Purdue University's Ray W. Herrick Laboratories. The Viper will be installed in a CO_2 test stand and the five operating conditions will be simulated. The power output from the Viper Expander will be compared to the theoretical and the device efficiency will be determined.

NOMENCLATURE

A	area	(m^3)
H	specific enthalpy	(kJ/kg)
$\dot{m}$	mass flow rate	(kg/s)
P	pressure	(kPa)
s	specific entropy	(kJ/kg-K)
$\dot{W}$	power	(W)
V	velocity	(m/s)

Greek symbols

Η	efficiency
Ρ	density
Ξ	improvement to COP

Subscript

1,2,3…	state points
C	cooling
In	inlet
Is	isentropic
Net	net total value
Out	outlet
Ref	refrigerant
V	Viper Expander
Viper	Viper Expander

Acronyms

AC	Alternating Current
DC	Direct Current
CFC	Chlorofluorocarbon
COP	Coefficient of Performance
ECM	Electronically Commutated Motor
ED-WOW	Expansion Device - With Output Work
EXV	Electronic Expansion Valve
GWP	Global Warming Potential
HFC	Hydrofluorocarbon
TXV	Thermostatic Expansion Valve
OP	Operating Point

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